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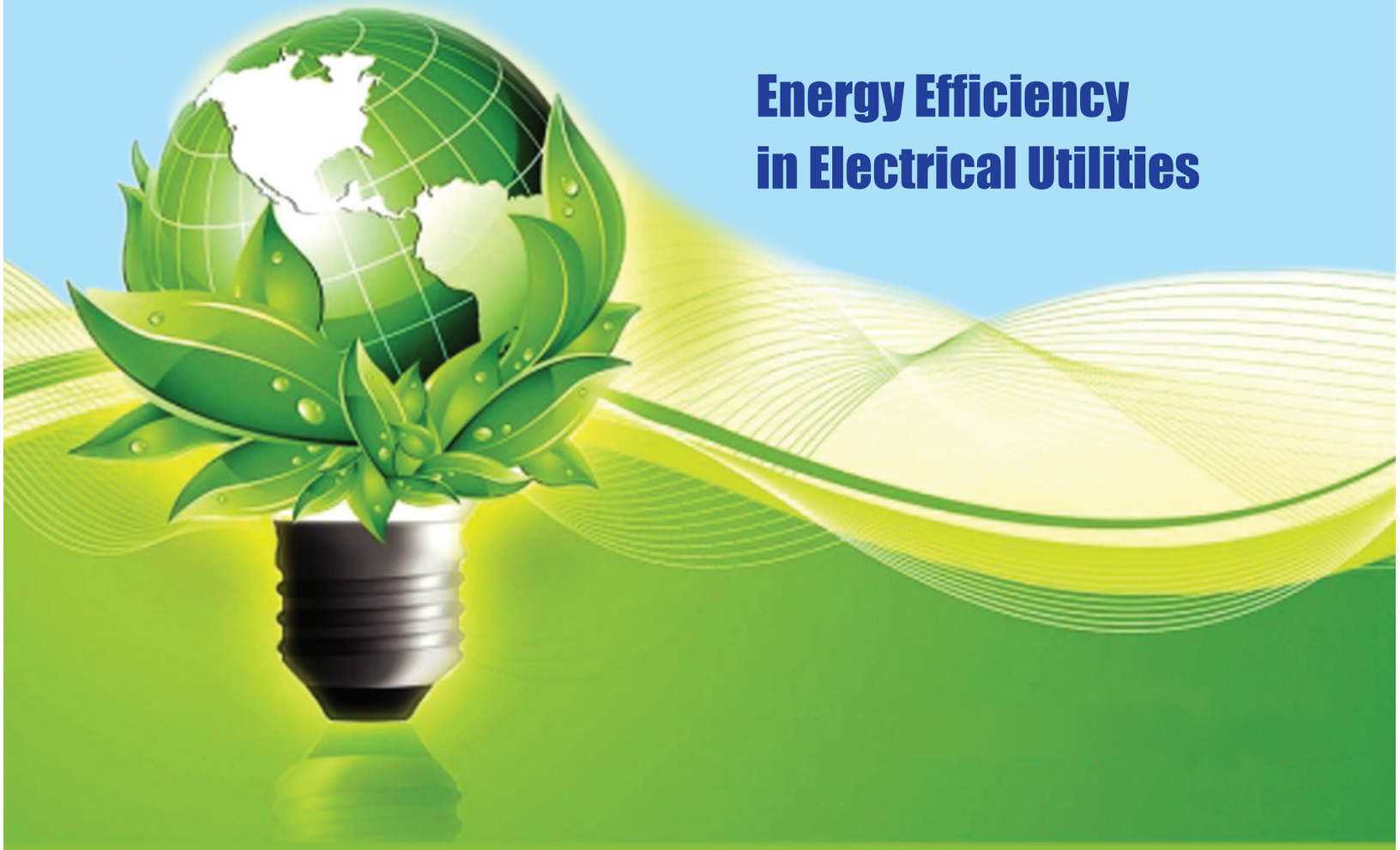

Energy Efficiency in Electrical Utilities

GUIDE BOOK

for

NATIONAL CERTIFICATION EXAMINATION

FOR ENERGY MANAGERS AND ENERGY AUDITORS

Bureau of Energy Efficiency

(A Statutory Body under Ministry of Power, Government of India)

4th Floor, Sewa Bhawan, R. K.Puram, New Delhi - 110 066

www.bee-india.nic.in

FOURTH EDITION, 2015

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PREFACE

National Certification Examination for Energy Managers and Energy Auditors aims to develop qualified professionals in the field of energy conservation in India.

At the initiative of Bureau of Energy Efficiency (BEE) for meeting the requirements of National Certification Examination for Energy Managers and Energy Auditors, Dr.Ambedkar Institute of Productivity, Chennai, the training institute of National Productivity Council of India has prepared four guide books in respect of four examinations as follows:

Guide Book	Examination Paper	Topic
Book-1	Paper-1	General Aspects of Energy Management and Energy Audit
Book-2	Paper-2	Energy Efficiency in Thermal Utilities
Book-3	**Paper-3**	**Energy Efficiency in Electrical Utilities**
Book-4	Paper-4	Energy Performance Assessment for Equipment and Utility Systems

This book-3 on Energy Efficiency in Electrical Utilities focuses on basic principles, guide to efficient operations, performance evaluation techniques of electrical energy systems and equipment such as motors, compressors, refrigeration systems, pumps, fans and blowers, cooling towers, DG sets and lighting system. It also contains guidelines for efficient use of electrical energy.

Electrical energy, constitute a significant portion of the energy bill in many industries, and offers considerable potential for energy savings.

These books have been prepared with material support from the various books referenced at the end of each chapter. We trust that these books, besides its intended purpose, will also help in enhancing the knowledge base of practicing engineers and managers in understanding and applying energy efficiency techniques.

We hope that this revised edition, incorporating recent developments and practices, will be of use to the prospective energy auditors, managers and other relevant energy efficiency professionals. We invite your valued suggestions and comments, which may be mailed to **guidebooks@em-ea.org**

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March 17, 2015

MESSAGE

India's economic growth depends on the long-term availability of energy from sources that are affordable, accessible and environmentally friendly. Increasing demands for fossil fuels have led to over exploitation of natural resources with adverse impacts on environment and the resultant health issues.

Energy efficiency and its conservation is one of the most cost effective measures to bridge the gap between demand and supply, and to mitigate the environmental impacts of rapid development.

It is heartening to note that the guide books prepared for the National Certification Examination for energy managers and auditors has caught the attention of a wide cross section of the engineering fraternity, not only in India but also in many countries around the world. Many of the techniques and technologies enumerated in these guide books towards enhancing energy efficiency have been put to practice and rich dividends have been realized. Thus these books besides meeting the requirements of the examination are also serving as a manual for practicing engineers.

I commend the efforts put in to prepare such a practical and comprehensive compilation. My best wishes to the users of these books.

(Pradeep K.Sinha)

ENERGY IS LIFE
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CONSERVE IT

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(भारत सरकार, विद्युत मंत्रालय)

BUREAU OF ENERGY EFFICIENCY
(Government of India, Ministry of Power)

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MESSAGE

The guide books for National Certification Examination for Energy Managers and Energy Auditors are an enthusiastic celebration of a wide variety of topics related to energy efficiency, especially those which are of practical nature, intricate structure and timely relevance. A glance through the pages of this book will show that it is an unusual type of publication. Though primarily intended for the purpose of being a study material for the examination these guide books have been found to be an out and out a practical guide to implementing energy savings measures in industry and establishments. That they have also been adopted by the energy efficiency promotion organisations of other countries is a testimony to the efficacy of these guide books. Also many of the Indian universities have adopted these books for Post Graduate programmes in energy management.

Written by a group of experienced professionals practicing energy efficiency the books explore the depth and breadth of various possibilities for saving energy in a wide gamut of processes and equipment. Guides such as these are no replacement for the necessary hard work required to realize the potential of energy efficiency but they can certainly help you ensure that your efforts are focused and effective.

I wish all the readers the very best.

(Ajay Mathur)

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ACKNOWLEDGEMENT

Energy efficiency has assumed a critical role in our pursuit of sustainable development. The gains from energy efficiency have been harnessed the world over and India is no exception. Information from the industry and other establishments reveal the rapid strides made in the area of energy efficiency in our country. Examples of industries which have achieved this potential are projected in the energy conservation award programme of Government of India.

As the low hanging fruits appear to have been picked up the path to success in energy management is getting steeper and steeper. More and more information, innovations and knowledge are required to further exploit the energy conservation potential. The continuous efforts to upgrade and refine the guide books for National Certification Examination for Energy Managers and Energy Auditors are aimed at equipping the reader with the resources to identify and implement various energy conservation measures. Through the system of examination we have been able to build sufficient skilled manpower in the area of energy efficiency. We are pleasantly surprised by the enormous interest evinced in the examination from all corners

The unwavering support for the development of these guide books and constructive comments from various quarters about their contents, have been invaluable.

(Bhaskar J. Sarma)

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Chapter – 3 Compressed Air System

Chapter – 4 HVAC and Refrigeration System

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Chapter 1

Electrical Systems

Learning Objectives

In this chapter you will learn about

- The electrical power supply links from generation to end-use
- The utility tariff structure, billing components and scope areas for bill reduction at plant/facility level
- Concept of maximum demand, metering, load profile analysis, maximum demand reduction through load factor improvements
- Basics of power factor, its importance, capacitor selection, sizing, location, benefits and automatic PF controller
- Basics of transformer, types and losses
- Factors affecting distribution losses and improvement option
- Assessment of T&D efficiency
- Standards and labeling programme for distribution transformer
- Harmonics and its effect on electrical systems

1. ELECTRICAL SYSTEM

Electricity billing, Electrical load management and maximum demand control, Power factor improvement and its benefit, Automatic power factor controllers, Selection and location of capacitors, Performance assessment of PF capacitors, Distribution and transformer losses, Energy efficient transformers, Standards & labeling programme of distribution transformers, Assessment of transmission and distribution efficiency, Demand side management, Losses due to harmonics and voltage unbalance.

1.1 Introduction to Electric Power Supply Systems

Electric power supply system comprises of generating units that produce electricity; high voltage transmission lines that transport electricity over long distances; distribution lines that deliver the electricity to consumers; substations that connect the pieces to each other; and energy control centers to coordinate the operation of the components.

The Figure 1.1 shows a simple electric supply system with Generating Station, Power transmission and distribution network and linkages from electricity sources to end-user.

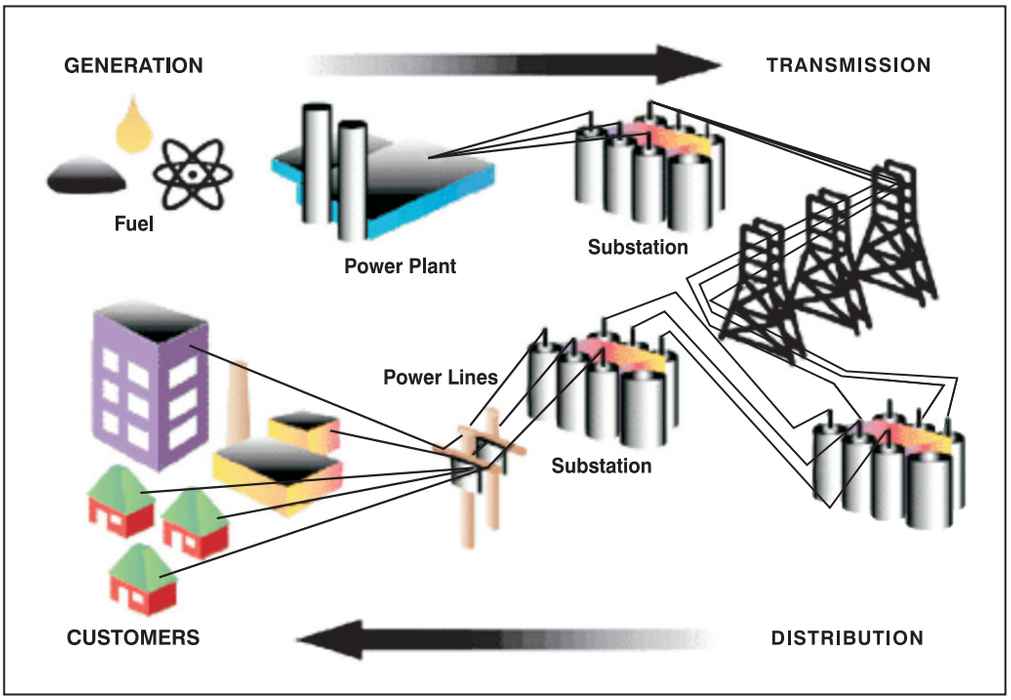


Figure 1.1 Typical Electric Power Supply Systems

Power Generation Plant

The fossil fuels such as coal, oil and natural gas, nuclear energy, and falling water (hydel) are commonly used energy sources in the power generating plant. A wide and growing variety of unconventional generation technologies and fuels have also been developed, including cogeneration, solar energy, wind generators, and waste materials.

About 70 % of power generating capacity in India is from coal based thermal power plants. The principle of coal-fired power generation plant is shown in Figure 1.2. Energy stored in the coal is converted in to electricity in a thermal power plant. Coal is pulverized to the consistency of talcum powder. Then powdered coal is blown into the water wall boiler where it is burned at temperature higher than 1300°C. The heat in the combustion gas is transferred into steam. This high-pressure steam is used to spin the steam turbine. Finally turbine rotates the generator to produce electricity.

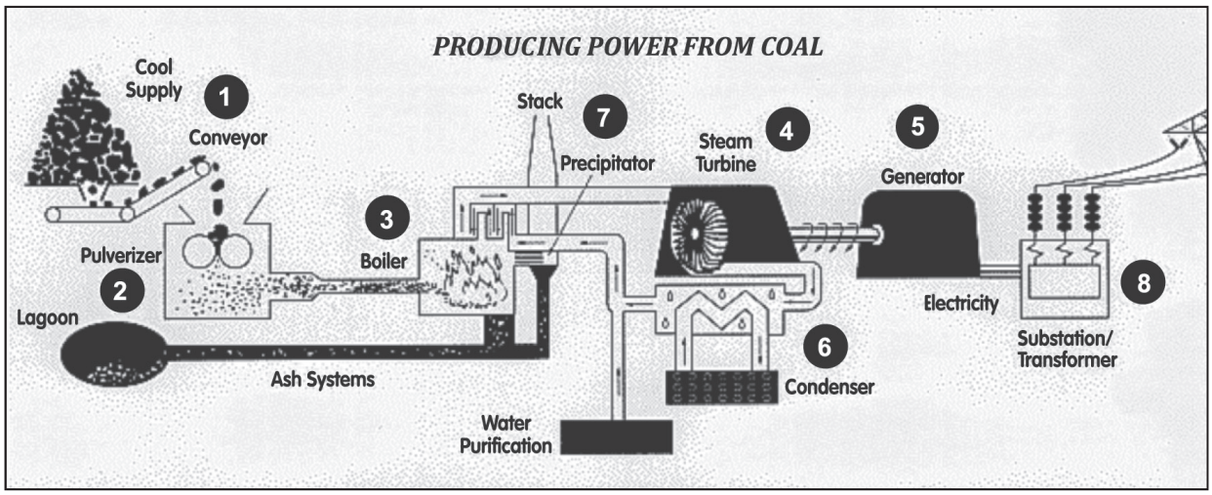


Figure 1.2 Principle of Thermal Power Generation

In India, for the coal based power plants, the overall efficiency ranges from 28% to 35% depending upon the size, operational practices, fuel quality and capacity utilization. Where fuels are the source of generation, a common term used is the "HEAT RATE" which reflects the efficiency of generation. "HEAT RATE" is the heat input in kilo Calories or kilo Joules, for generating 'one' kilo Watt-hour of electrical output. One kilo Watt hour of electrical energy being equivalent to 860 kilo Calories of thermal energy or 3600 kilo Joules of thermal energy. The "HEAT RATE" is inversely proportional to efficiency of power generation i.e., lower the heat rate, higher is the generation efficiency.

Transmission and Distribution Lines:

The power plants typically produce 50 cycle/second (Hertz), alternating-current (AC) electricity with voltages between 11kV and 33kV. At the power plant site, the 3-phase voltage is stepped up to a higher voltage for transmission on cables strung on cross-country towers.

High voltage (HV) and extra high voltage (EHV) transmission is the next stage from power plant to transport A.C. power over long distances at voltages like; 220 kV & 400 kV (Figure 1.3). Where transmission is over 1000 kM, high voltage direct current transmission is also favored to minimize the losses.

Sub-transmission network at 132 kV, 110 kV, 66 kV or 33 kV constitutes the next link towards the end user. Distribution at 11

Figure 1.3
High Voltage Transmission Lines

kV / 6.6 kV / 3.3 kV constitutes the last link to the consumer, who is connected directly or through transformers depending upon the drawn level of service. The transmission and distribution network include sub-stations, lines and distribution transformers. High voltage transmission is used so that smaller, more economical wire sizes can be employed to carry the lower current and to reduce losses. Sub-stations, containing step-down transformers, reduce the voltage for distribution to industrial users. The voltage is further reduced for commercial facilities. Electricity must be generated, as and when it is needed since electricity cannot be stored virtually in the system. Typical voltage levels in a power system are given in Figure 1.4.

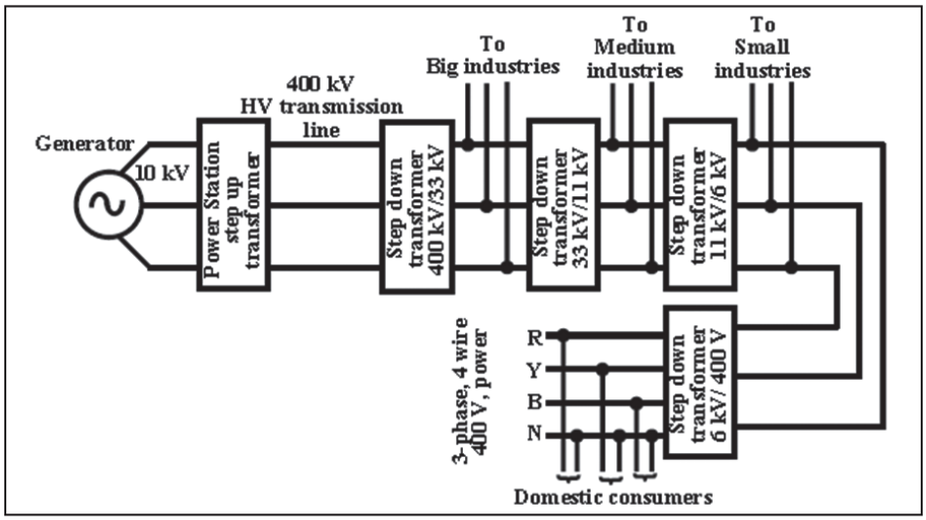


Figure 1.4 Typical Voltage Levels in a Power System

There is no difference between a transmission line and a distribution line except for the voltage level and power handling capability. Transmission lines are usually capable of transmitting large quantities of electric energy over great distances. They operate at high voltages. Distribution lines carry limited quantities of power over shorter distances.

Voltage drops in line are in relation to the resistance and reactance of line, length and the current drawn. For the same quantity of power handled, lower the voltage, higher the current drawn and higher the voltage drop. The current drawn is inversely proportional to the voltage level for the same quantity of power handled.

The power loss in line is proportional to resistance and square of current. (i.e. $P_{Loss}=I^2R$). Higher voltage transmission and distribution thus would help to minimize line voltage drop in the ratio of voltages, and the line power loss in the ratio of square of voltages. For instance, if distribution of power is raised from 11 kV to 33 kV, the voltage drop would be lower by a factor 1/3 and the line loss would be lower by a factor $(1/3)^2$ i.e., 1/9. Lower voltage transmission and distribution also calls for bigger size conductor on account of current handling capacity needed.

Cascade Efficiency

The primary function of transmission and distribution equipment is to transfer power economically and reliably from one location to another.

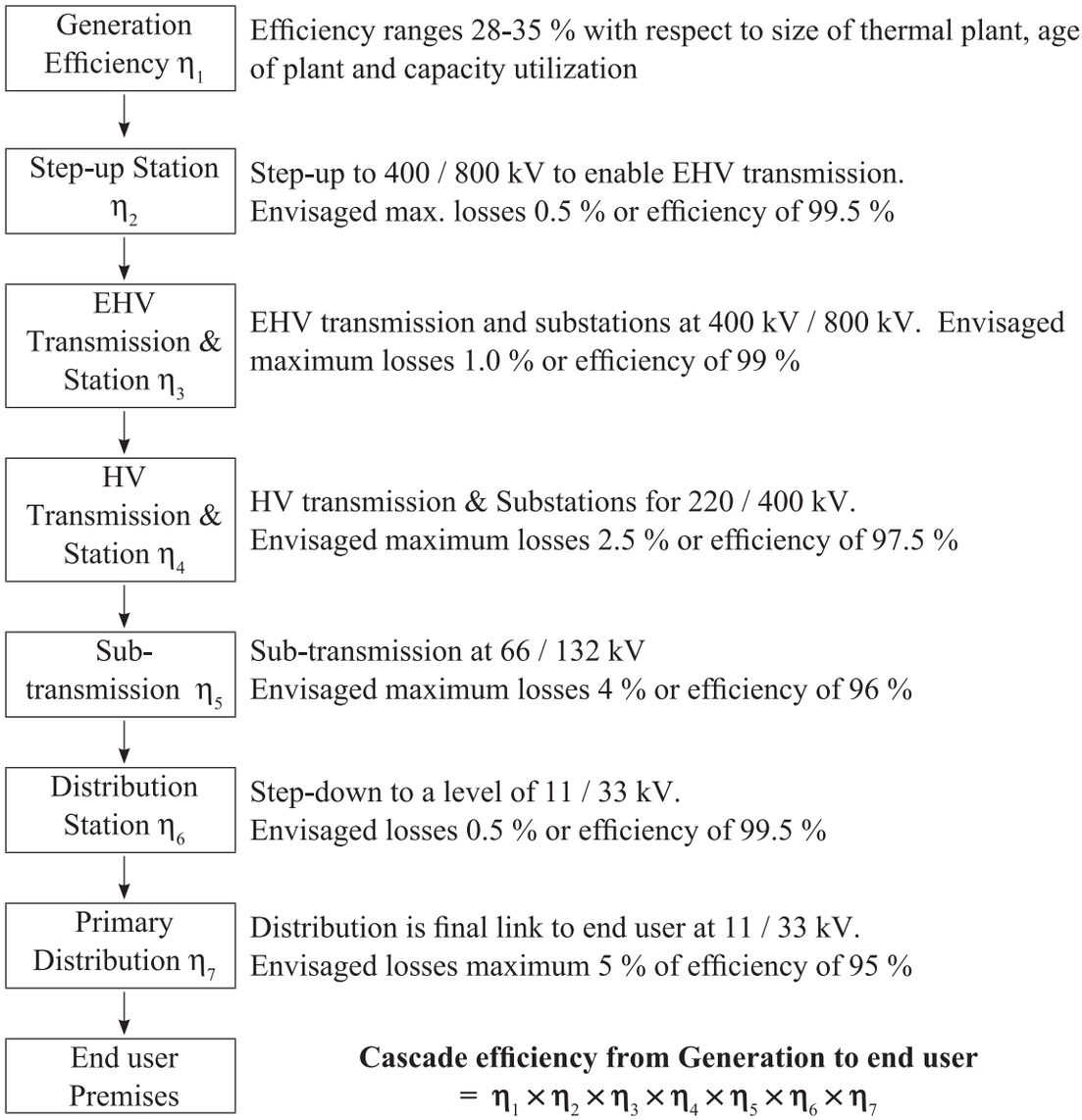
Conductors in the form of wires and cables strung on towers and poles carry the high-voltage, AC electric current. A large number of copper or aluminum conductors are used to form the transmission path. The resistance of the long-distance transmission conductors is to be minimized. Energy loss in transmission lines is wasted in the form of I^2R losses.

Capacitors are used to correct power factor by causing the current to lead the voltage. When the AC currents are kept in phase with the voltage, operating efficiency of the system is maintained at a high level.

Circuit-interrupting devices are switches, relays, circuit breakers, and fuses. Each of these devices is designed to carry and interrupt certain levels of current. Making and breaking the current carrying conductors in the transmission path with a minimum of arcing is one of the most important characteristics of this device. Relays sense abnormal voltages, currents, and frequency and operate to protect the system.

Transformers are placed at strategic locations throughout the system to minimize power losses in the T&D system. They are used to change the voltage level from low-to-high in step-up transformers and from high-to-low in step-down units.

The power source to end user energy efficiency link is a key factor, which influences the energy input at the source of supply. If we consider the electricity flow from generation to the user in terms of cascade energy efficiency, typical cascade efficiency profile from generation to 11 – 33 kV user industry will be as follows:



Cascade efficiency from Generation to end user

$$= \eta_1 \times \eta_2 \times \eta_3 \times \eta_4 \times \eta_5 \times \eta_6 \times \eta_7$$

The cascade efficiency in the T&D system from output of the power plant to the end use is 87% (i.e. 0.995 x 0.99 x 0.975 x 0.96 x 0.995 x 0.95 = 87%)

After power generation at the plant it is transmitted and distributed over a wide network. The standard technical losses are around 17 % in India (Efficiency=83%). But the figures for many of the states show T & D losses ranging from 17 – 50 %. All these may not constitute technical losses, since un-metered and pilferage are also accounted in this loss.

Industrial End User

At the industrial end user premises, again the plant network elements like transformers at receiving sub-station, switchgear, lines and cables, load-break switches, capacitors cause losses, which affect the input-received energy. However the losses in such systems are meager and unavoidable.

A typical plant single line diagram of electrical distribution system is shown in Figure 1.5

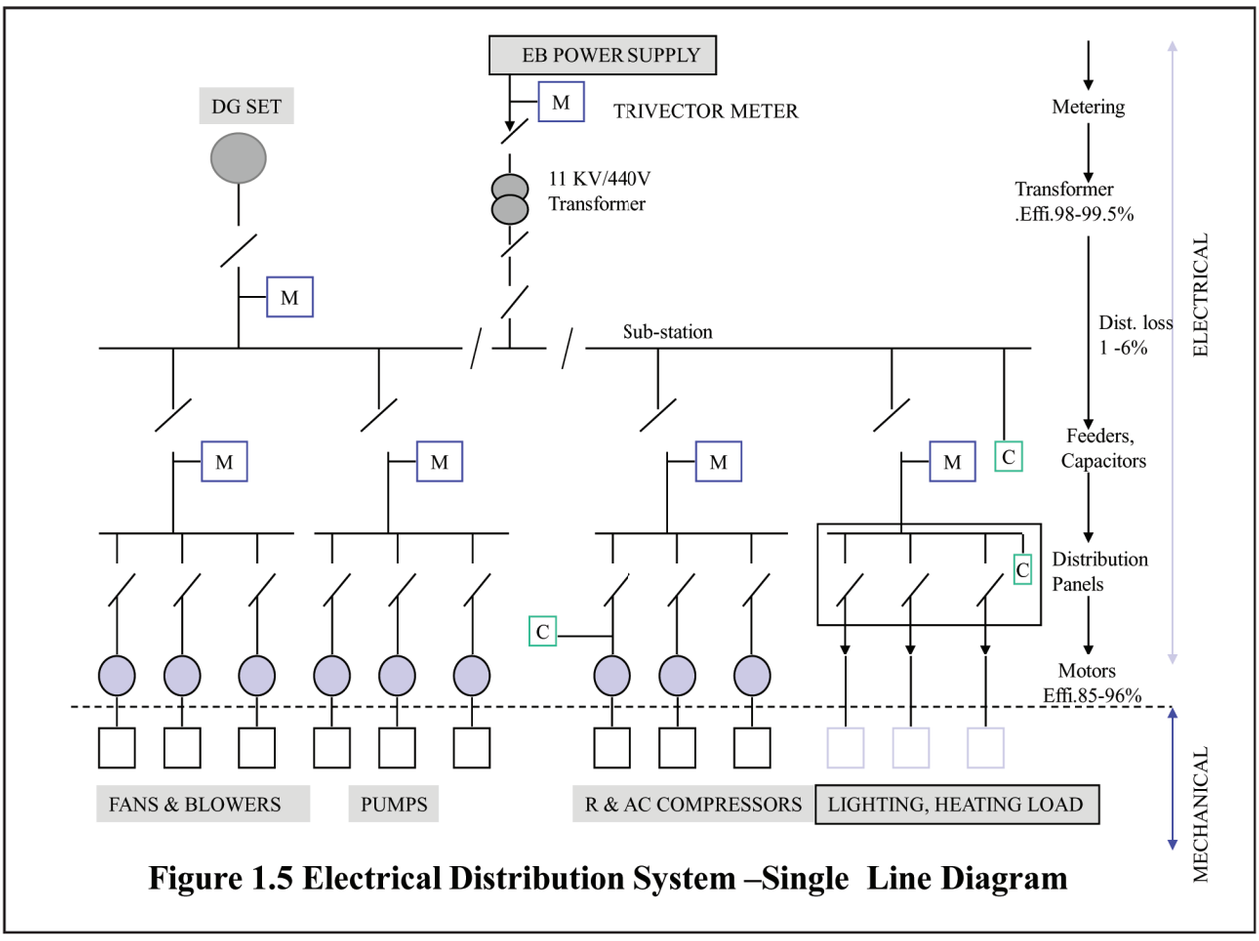


Figure 1.5 Electrical Distribution System –Single Line Diagram

ONE Unit saved = TWO Units Generated

When the power reaches the industry, it meets the transformer. The energy efficiency of the transformer is generally very high. Next, it goes to the motor through internal plant distribution network. A typical distribution network efficiency including transformer is 95% and motor efficiency is about 90%.

Another 30 % (Efficiency=70%) is lost in the mechanical system which includes coupling/ drive train, a driven equipment such as pump and flow control valves/throttling etc. Thus the overall energy efficiency becomes 50%. (0.83 x 0.95x 0.9 x 0.70 = 0.50, i.e. 50% efficiency)

Hence one unit saved in the end user is equivalent to two units generated in the power plant. (1Unit / 0.5Eff = 2 Units)

1.2 Electricity Billing

The electricity billing by utilities for medium & large enterprises, in High Tension (HT) category, is often done on two-part tariff structure, i.e. one part for capacity (or demand) drawn and the second part for actual energy drawn during the billing cycle. Capacity or demand is in kVA (apparent power) or kW terms. The reactive energy (i.e.) kVArh drawn by the service is also recorded and billed for in some utilities, because this would affect the load on the utility. Accordingly, utility charges for maximum demand, active energy and reactive power drawn (as reflected by the power factor) in its billing structure. In addition, other fixed and variable expenses are also levied.

The tariff structure generally includes the following components:

a) *Maximum demand Charges*

These charges relate to maximum demand registered during month/billing period and corresponding rate of utility.

b) *Energy Charges*

These charges relate to energy (kilowatt hours) consumed during month / billing period and corresponding rates, often levied in slabs of use rates. Some utilities now charge on the basis of apparent energy (kVAh), which is a vector sum of kWh and kVArh.

c) *Power factor* penalty or bonus rates, as levied by most utilities, are to contain reactive power drawn from grid.

d) *Fuel cost* adjustment charges as levied by some utilities are to adjust the increasing fuel expenses over a base reference value.

e) *Electricity duty charges* levied with respect to units consumed.

f) *Meter rentals*

g) *Lighting and fan power consumption* is often at higher rates, levied sometimes on slab basis or on actual metering basis.

h) *Time of Day (TOD) rates* like peak and non-peak hours are also prevalent in tariff structure provisions of some utilities.

i) *Penalty for exceeding contract demand*

j) *Surcharge if metering is at LT side in some of the utilities*

Analysis of utility bill data and monitoring its trends helps energy manager to identify ways for electricity bill reduction through available provisions in tariff framework, apart from energy budgeting.

The utility employs an electromagnetic or electronic trivector meter, for billing purposes. The minimum outputs from the electromagnetic meters are:

- Maximum demand registered during the month, which is measured in preset time intervals (say of 30 minute duration) and this is reset at the end of every billing cycle.
- Active energy in kWh during billing cycle
- Reactive energy in kVArh during billing cycle and
- Apparent energy in kVAh during billing cycle

It is important to note that while maximum demand is recorded, it is not the instantaneous demand drawn, as is often misunderstood, but the time integrated demand over the predefined recording cycle.

As an example, in an industry, if the drawl over a recording cycle of 30 minutes is:

2500 kVA for 4 minutes
3600 kVA for 12 minutes
4100 kVA for 6 minutes
3800 kVA for 8 minutes

The MD recorder will be computing MD as:

$$\frac{(2500x4) + (3600 \times 12) + (4100 \times 6) + (3800 \times 8)}{30} = 3606.7 \text{ kVA}$$

The month's maximum demand will be the highest among such demand values recorded over the month. The meter registers only if the value exceeds the previous maximum demand value and thus, even if, average maximum demand is low, the industry / facility has to pay for the maximum demand charges for the highest value registered during the month, even if it occurs for just one recording cycle duration i.e., 30 minutes during whole of the month. A typical demand curve is shown in Figure 1.6.

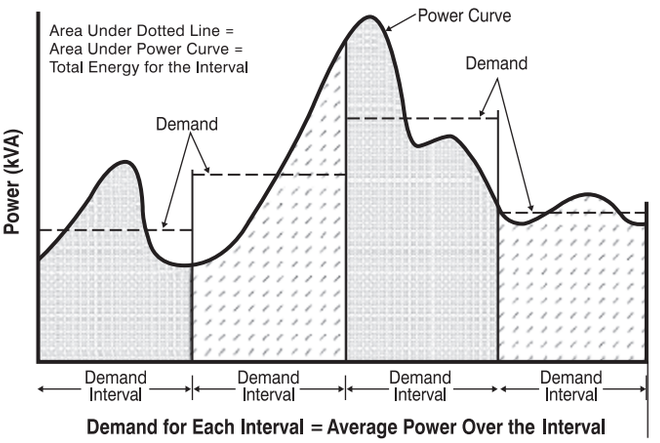


Figure 1.6 Demand Curve

As can be seen from the Figure 1.6 above the demand varies from time to time. The demand is measured over predetermined time interval and averaged out for that interval as shown by the horizontal dotted line.

Of late most electricity boards have changed over from conventional electromechanical trivector meters to electronic meters, which have some excellent provisions that can help the utility as well as the industry. These provisions include:

- Substantial memory for logging and recording all relevant events
- High accuracy up to 0.2 class

- Amenability to time of day tariffs
- Tamper detection /recording
- Measurement of harmonics and Total Harmonic Distortion (THD)
- Long service life due to absence of moving parts
- Amenability for remote data access/downloads

Trend analysis of purchased electricity and cost components can help the industry to identify key result areas for bill reduction within the utility tariff available framework in Table 1.1.

Table 1.1 Purchased Electrical Energy Trend

Month & Year	MD Recorded kVA	Billing Demand* kVA	Total Energy Consumption kWh	Energy Consumption During Peak Hours (kWh)	MD Charge Rs./kVA	Energy Charge Rs./kWh	PF	PF Penalty /Rebate Rs.	Total Bills Rs.	Average Cost Rs./kWh
Jan.										
Feb.										
…….										
…….										
Dec.										

*Some utilities charge Maximum Demand on the basis of minimum billing demand, which may be between 75 to 100% of the contract demand or actual recorded demand whichever is higher

1.3 Electrical Load Management and Maximum Demand Control

Need for Electrical Load Management

In a macro perspective, the growth in the electricity use and diversity of end use segments in time of use has led to shortfalls in capacity to meet demand. As capacity addition is costly and only a long time prospect, better load management at user end helps to minimize peak demands on the utility infrastructure as well as better utilization of power plant capacities.

The utilities (Distribution companies) use power tariff structure to influence end user in better load management through measures like time of use tariffs, penalties on exceeding allowed maximum demand, night tariff concessions etc. Load management is a powerful means of efficiency improvement both for end user as well as utility.

As the demand charges constitute a considerable portion of the electricity bill, from user angle too there is a need for integrated load management to effectively control the maximum demand.

Step By Step Approach for Maximum Demand Control

1. Load Curve Generation

Presenting the load demand of a consumer against time of the day is known as a ‘load curve’. If it is plotted for the 24 hours of a single day, it is known as an ‘hourly load curve’ and if daily demands plotted over a month, it is called ‘daily load curve’. A

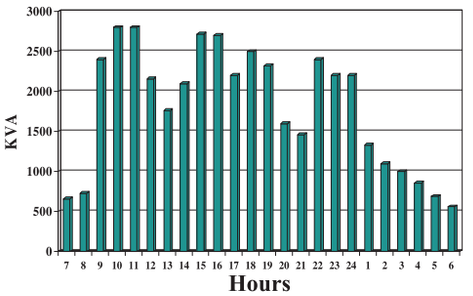


Figure 1.7 Maximum Demand (Daily Load Curve, Hourly kVA)

typical hourly load curve for an engineering industry is shown in Figure 1.7. These types of curves are useful in predicting patterns of drawl, peaks and valleys and energy use trend in a section or in an industry or in a distribution network as the case may be.

2. Rescheduling of Loads

Rescheduling of large electric loads and equipment operations, in different shifts can be planned and implemented to minimize the simultaneous maximum demand. For this purpose, it is advisable to prepare an operation flow chart and a process chart. Analyzing these charts and with an integrated approach, it would be possible to reschedule the operations and running equipment in such a way as to improve the load factor which in turn reduces the maximum demand.

3. Storage of Products/in process material/ process utilities like refrigeration

It is possible to reduce the maximum demand by building up storage capacity of products/ materials, water, chilled water / hot water, using electricity during off peak periods. Off peak hour operations also help to save energy due to favorable conditions such as lower ambient temperature etc.

Example: Ice bank system is used in milk & dairy industry. Ice is made in lean period and used in peak load period and thus maximum demand is reduced.

4. Shedding of Non-Essential Loads

When the maximum demand tends to reach preset limit, shedding some of non-essential loads temporarily can help to reduce it. It is possible to install direct demand monitoring and control systems (Figure 1.8), which will switch off non-essential loads when a preset demand is reached. Simple systems give an alarm, and the loads are shed manually. Sophisticated microprocessor controlled systems are also available, which provide a wide variety of control options like:

- Accurate prediction of demand
- Graphical display of present load, available load, demand limit
- Visual and audible alarm
- Automatic load shedding in a predetermined sequence
- Automatic restoration of load
- Recording and metering

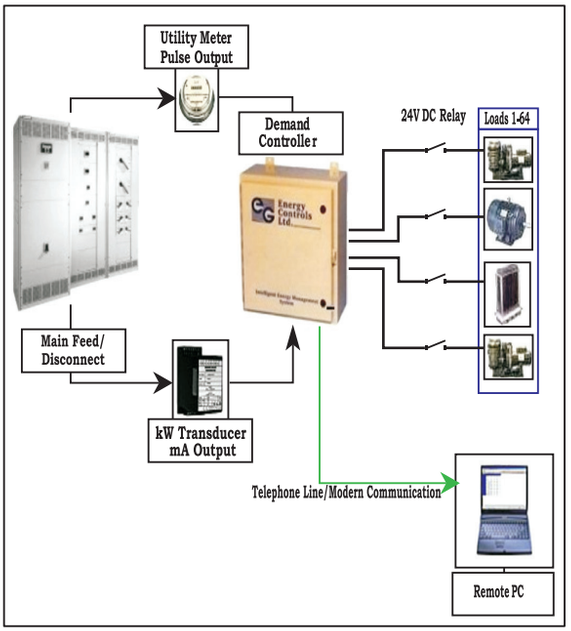


Figure 1.8 Maximum Demand Controller

5. Operation of Captive Generation and Diesel Generation Sets

When diesel generation sets are used to supplement the power supplied by the electric utilities, it is advisable to connect the D.G. sets for durations when demand reaches the peak value. This would reduce the load demand to a considerable extent and minimize the demand charges.

6. Reactive Power Compensation

The maximum demand can also be reduced at the plant level by using capacitor banks and maintaining the optimum power factor. Capacitor banks are available with microprocessor based control systems. These systems switch on and off the capacitor banks to maintain the desired Power factor of system and optimize maximum demand thereby.

1.4 Power Factor Improvement and Benefits

Power factor Basics

In all industrial electrical distribution systems, the major loads are resistive and inductive. Resistive loads are incandescent lighting and resistance heating. In case of pure resistive loads, the voltage (V),current (I), resistance (R) relations are linearly related, i.e.

$$V = I \times R \text{ and Power (kW)} = V \times I$$

Typical inductive loads are A.C. Motors, induction furnaces, transformers and ballast-type lighting. Inductive loads require two kinds of power: a) active (or working) power to perform the work and b) reactive power to create and maintain electro-magnetic fields.

Active power is measured in kW (Kilo Watts). Reactive power is measured in kVAr (kilo Volt-Amperes Reactive).

The vector sum of the active power and reactive power make up the total (or apparent) power used. This is the power generated by the SEBs for the user to perform a given amount of work. Total Power is measured in kVA (kilo Volts-Amperes) (See Figure 1.9).

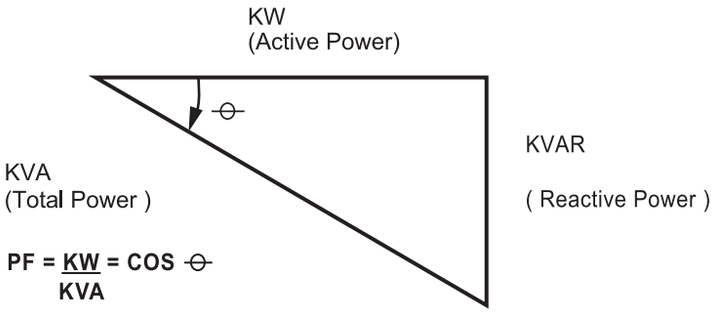


Figure 1.9 kW, kVAr and kVA Vector

The active power (shaft power required or true power required) in kW and the reactive power required (kVAr) are 90^0 apart vectorically in a pure inductive circuit i.e., reactive power kVAr lagging the active kW. The vector sum of the two is called the apparent power or kVA, as illustrated above and the kVA reflects the actual electrical load on distribution system.

The ratio of kW to kVA is called the power factor, which is always less than or equal to unity. Theoretically, when electric utilities supply power, if all loads have unity power factor, maximum power can be transferred for the same distribution system capacity. However, as the loads are inductive in nature, with the power factor ranging from 0.2 to 0.9, the electrical distribution network is stressed for capacity at low power factors.

Improving Power Factor

Figure 1.10 Capacitors

The solution to improve the power factor is to add power factor correction capacitors (see Figure 1.10) to the plant power distribution system. They act as reactive power generators, and provide the needed reactive power to accomplish kW of work. This reduces the amount of reactive power, and thus total power, generated by the utilities.

Example:

A chemical industry had installed a 1500 kVA transformer. The initial demand of the plant was 1160 kVA with power factor of 0.70. The % loading of transformer was about 78% (1160/1500 = 77.3%). To improve the power factor and to avoid the penalty, the unit had added about 410 kVAr in motor load end. This improved the power factor to 0.89, and reduced the required kVA to 913, which is the vector sum of kW and kVAr (see Figure 1.11).

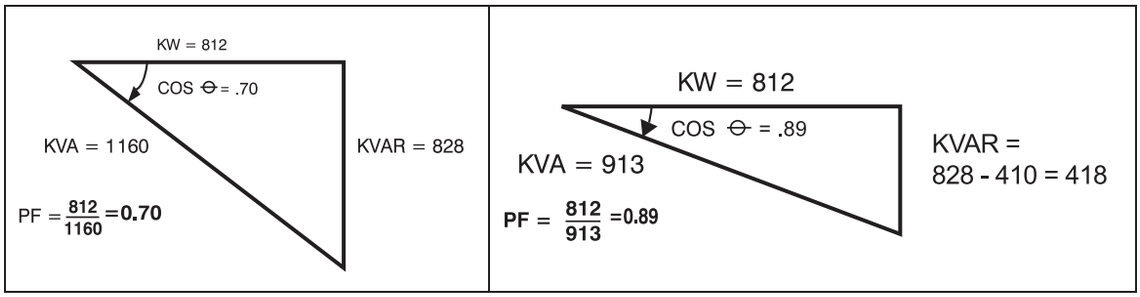


Figure 1.11 Power factor before and after Improvement

After improvement the plant has avoided penalty and the 1500 kVA transformer is now loaded only to 60% of capacity. This will allow the addition of more loads in the future to be supplied by the transformer.

The advantages of PF improvement by capacitor addition

a) Reactive component of the network is reduced and so also the total current in the system from the source end.

b) I^2R power losses are reduced in the system because of reduction in current.

c) Voltage level at the load end is increased.

d) kVA loading on the source generators as also on the transformers and lines up to the capacitors reduces giving capacity relief. A high power factor can help in utilizing the full capacity of the electrical system.

Cost benefits of PF improvement

While costs of PF improvement are in terms of investment needs for capacitor addition the benefits to be quantified for feasibility analysis are:

a) Reduced kVA (Maximum demand) charges in utility bill

b) Reduced distribution losses (KWH) within the plant network

c) Better voltage at motor terminals and improved performance of motors

d) A high power factor eliminates penalty charges imposed when operating with a low power factor

e) Investment on system facilities such as transformers, cables, switchgears etc for delivering load is reduced.

Automatic Power Factor Controllers

Many of the industries desire to maintain the power factor near unity with the objective of minimizing the maximum demand as well as availing the PF incentives offered by DISCOM's. When the loads in the industries are fluctuating it becomes difficult to maintain near unity PF with fixed capacitor banks. At low loads there is a possibility of PF going into leading side which can create high voltages at the motor terminals. In such cases the maximum demand will also rise. To overcome this situation automatic power factor controllers are deployed.

Power factor controllers are typically panel mount and used like a panel mount meter, indicating the power factor at the point of supply (Figure 1.12). Power factor controllers are programmable and range from quite simple to very complex.

A simple power factor controller monitors the displacement power factor. The controller displays the power factor on a digital display and compares the measured power factor with the desired power factor. If the power factor is less than the desired power factor, another bank of capacitors is switched on via a relay output on the controller. If the power factor is leading, or is above a threshold point, a bank of capacitors is switched OFF.

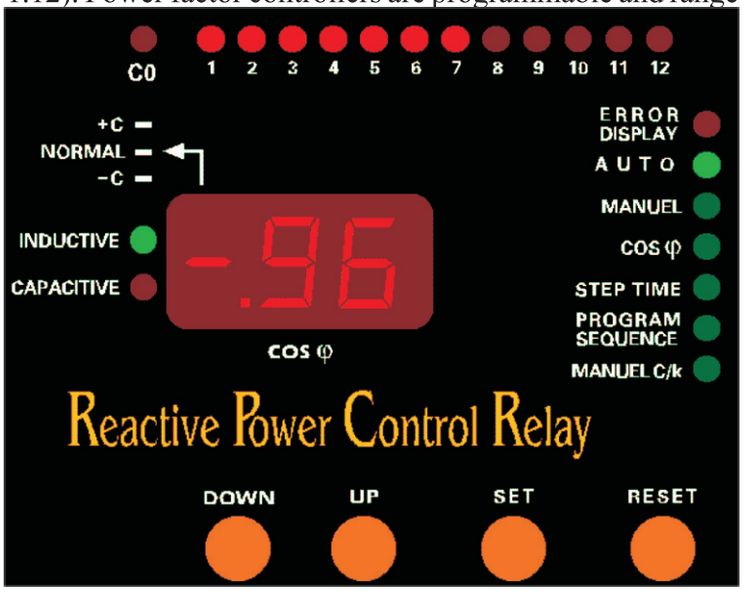


Figure 1.12 Automatice Power Factor Control Relay

The controller has a number of relay outputs for controlling contactors switching capacitors. Typically, the number of outputs will range from 6 to 14 relays. The number and size of the banks being switched is dependent on the type of load, the range of control required and the designated power factor range. Some controllers expect that equal stages will be used, and others are quite flexible. The top end controllers measure the size of each step and calculate which step combinations will give the best results. In this case, it is possible to use a combination of step sizes. A good configuration is to use at least two small steps and at least four large steps. For large installations, up to 14 stages can be used. The number of times that a bank can be switched is limited with delay ON and delay OFF times that are programmable. Some controllers keep the numbers of operations equal across all banks. There are

a number of other options that can be included such as harmonic current alarms and low current thresholds to prevent capacitors being connected under very light load.

Selection and location of capacitors

The capacitors can be selected based on the following relation

$$\text{kVAr Rating} = \text{kW} [\tan \phi 1 - \tan \phi 2]$$

Where, kVAr rating is the size of the capacitor needed, kW is the average power drawn, tan $\phi 1$ is the trigonometric ratio for the present power factor, and tan ϕ_2 is the trigonometric ratio for the desired PF.

$$\phi_1 = \text{Existing } (\text{Cos}^{-1} \text{ PF}_1) \text{ and } \quad \phi_2 = \text{Improved } (\text{Cos}^{-1} \text{ PF}_2)$$

Alternatively the Table 1.2 can be used for capacitor sizing.

The figures given in table are the multiplication factors which are to be multiplied with the input power (kW) to give the kVAr of capacitance required to improve present power factor to a new desired power factor.

Example:

The utility bill shows an average power factor of 0.72 with an average KW of 627. How much kVAr is required to improve the power factor to 0.95 ?

Using formula

$$\textit{Cos } \Phi 1 = 0.72 \text{ , } \textit{tan } \Phi 1 = 0.963$$

$$\textit{Cos } \Phi 2 = 0.95 \text{ , } \textit{tan } \Phi 2 = 0.329$$

$$\textit{kVAr required} = \text{P} (\tan\phi 1 - \tan\phi 2) = 627 (0.964 - 0.329)$$
$$= 398 \text{ kVAr}$$

Using table (see Table 1.2)

1) Locate 0.72 (original power factor) in column (1).
2) Read across desired power factor to 0.95 column. We find 0.635 multiplier
3) Multiply 627 (average kW) by 0.635 = 398 kVAr.
4) Install 400 kVAr to improve power factor to 95%.

Table 1.2 Multipliers to Determine Capacitor kVAr Requirements for Power Factor Correction

Original Power Factor	Desired Power Factor																					
	0.80	0.81	0.82	0.83	0.84	0.85	0.86	0.87	0.88	0.89	0.90	0.91	0.92	0.93	0.94	0.95	0.96	0.97	0.98	0.99	1.0
0.50	0.982	1.008	1.034	1.060	1.086	1.112	1.139	1.165	1.192	1.220	1.248	1.276	1.306	1.337	1.369	1.403	1.440	1.481	1.529	1.589	1.732
0.51	0.937	0.962	0.989	1.015	1.041	1.067	1.094	1.120	1.147	1.175	1.203	1.231	1.261	1.292	1.324	1.358	1.395	1.436	1.484	1.544	1.687
0.52	0.893	0.919	0.945	0.971	0.997	1.023	1.050	1.076	1.103	1.131	1.159	1.187	1.217	1.248	1.280	1.314	1.351	1.392	1.440	1.500	1.643
0.53	0.850	0.876	0.902	0.928	0.954	0.980	1.007	1.033	1.060	1.088	1.116	1.144	1.174	1.205	1.237	1.271	1.308	1.349	1.397	1.457	1.600
0.54	0.809	0.835	0.861	0.887	0.913	0.939	0.968	0.992	1.019	1.047	1.075	1.103	1.133	1.164	1.196	1.230	1.267	1.308	1.356	1.416	1.559
0.55	0.769	0.795	0.821	0.847	0.873	0.899	0.926	0.952	0.979	1.007	1.035	1.063	1.093	1.124	1.156	1.190	1.227	1.268	1.316	1.376	1.519
0.56	0.730	0.756	0.782	0.808	0.834	0.860	0.887	0.913	0.940	0.968	0.996	1.024	1.054	1.085	1.117	1.151	1.188	1.229	1.277	1.337	1.480
0.57	0.692	0.718	0.744	0.770	0.796	0.822	0.849	0.875	0.902	0.930	0.958	0.986	1.016	1.047	1.079	1.113	1.150	1.191	1.239	1.299	1.442
0.58	0.655	0.681	0.707	0.733	0.759	0.785	0.812	0.838	0.865	0.893	0.921	0.949	0.979	1.010	1.042	1.076	1.113	1.154	1.202	1.262	1.405
0.59	0.619	0.645	0.671	0.697	0.723	0.749	0.776	0.802	0.829	0.857	0.885	0.913	0.943	0.974	1.006	1.040	1.077	1.118	1.166	1.226	1.369
0.60	0.583	0.609	0.635	0.661	0.687	0.713	0.740	0.766	0.793	0.821	0.849	0.877	0.907	0.938	0.970	1.004	1.041	1.082	1.130	1.190	1.333
0.61	0.549	0.575	0.601	0.627	0.653	0.679	0.706	0.732	0.759	0.787	0.815	0.843	0.873	0.904	0.936	0.970	1.007	1.048	1.096	1.156	1.299
0.62	0.516	0.542	0.568	0.594	0.620	0.646	0.673	0.699	0.726	0.754	0.782	0.810	0.840	0.871	0.903	0.937	0.974	1.015	1.063	1.123	1.266
0.63	0.483	0.509	0.535	0.561	0.587	0.613	0.640	0.666	0.693	0.721	0.749	0.777	0.807	0.838	0.870	0.904	0.941	0.982	1.030	1.090	1.233
0.64	0.451	0.474	0.503	0.529	0.555	0.581	0.608	0.634	0.661	0.689	0.717	0.745	0.775	0.806	0.838	0.872	0.909	0.950	0.998	1.068	1.201
0.65	0.419	0.445	0.471	0.497	0.523	0.549	0.576	0.602	0.629	0.657	0.685	0.713	0.743	0.774	0.806	0.840	0.877	0.918	0.966	1.026	1.169
0.66	0.388	0.414	0.440	0.466	0.492	0.518	0.545	0.571	0.598	0.626	0.654	0.682	0.712	0.743	0.775	0.809	0.846	0.887	0.935	0.995	1.138
0.67	0.358	0.384	0.410	0.436	0.462	0.488	0.515	0.541	0.568	0.596	0.624	0.652	0.682	0.713	0.745	0.779	0.816	0.857	0.905	0.965	1.108
0.68	0.328	0.354	0.380	0.406	0.432	0.458	0.485	0.511	0.538	0.566	0.594	0.622	0.652	0.683	0.715	0.749	0.786	0.827	0.875	0.935	1.078
0.69	0.299	0.325	0.351	0.377	0.403	0.429	0.456	0.482	0.509	0.537	0.565	0.593	0.623	0.654	0.686	0.720	0.757	0.798	0.846	0.906	1.049
0.70	0.270	0.296	0.322	0.348	0.374	0.400	0.427	0.453	0.480	0.508	0.536	0.564	0.594	0.625	0.657	0.691	0.728	0.769	0.817	0.877	1.020
0.71	0.242	0.268	0.294	0.320	0.346	0.372	0.399	0.425	0.452	0.480	0.508	0.536	0.566	0.597	0.629	0.663	0.700	0.741	0.789	0.849	0.992
0.72	0.214	0.240	0.266	0.292	0.318	0.344	0.371	0.397	0.424	0.452	0.480	0.508	0.538	0.569	0.601	0.635	0.672	0.713	0.761	0.821	0.964
0.73	0.186	0.212	0.238	0.264	0.290	0.316	0.343	0.369	0.396	0.424	0.452	0.480	0.510	0.541	0.573	0.607	0.644	0.685	0.733	0.793	0.936
0.74	0.159	0.185	0.211	0.237	0.253	0.289	0.316	0.342	0.369	0.397	0.425	0.453	0.483	0.514	0.546	0.580	0.617	0.658	0.706	0.766	0.909
0.75	0.132	0.158	0.184	0.210	0.236	0.262	0.289	0.315	0.342	0.370	0.398	0.426	0.456	0.487	0.519	0.553	0.590	0.631	0.679	0.739	0.882
0.76	0.105	0.131	0.157	0.183	0.209	0.235	0.262	0.288	0.315	0.343	0.371	0.399	0.429	0.460	0.492	0.526	0.563	0.604	0.652	0.712	0.855
0.77	0.079	0.105	0.131	0.157	0.183	0.209	0.236	0.262	0.289	0.317	0.345	0.373	0.403	0.434	0.486	0.500	0.537	0.578	0.626	0.685	0.829
0.78	0.052	0.078	0.104	0.130	0.156	0.182	0.209	0.235	0.262	0.290	0.318	0.346	0.376	0.407	0.439	0.473	0.510	0.551	0.599	0.659	0.802
0.79	0.026	0.052	0.078	0.104	0.130	0.156	0.183	0.209	0.236	0.264	0.292	0.320	0.350	0.381	0.413	0.447	0.484	0.525	0.573	0.633	0.776
0.80	0.000	0.026	0.052	0.078	0.104	0.130	0.157	0.183	0.210	0.238	0.266	0.294	0.324	0.355	0.387	0.421	0.458	0.499	0.547	0.609	0.750
0.81		0.000	0.026	0.052	0.078	0.104	0.131	0.157	0.184	0.212	0.240	0.268	0.298	0.329	0.361	0.395	0.432	0.473	0.521	0.581	0.724
0.82			0.000	0.026	0.052	0.078	0.105	0.131	0.158	0.186	0.214	0.242	0.272	0.303	0.335	0.369	0.406	0.447	0.495	0.555	0.698
0.83				0.000	0.026	0.052	0.079	0.105	0.132	0.160	0.188	0.216	0.246	0.277	0.309	0.343	0.380	0.421	0.469	0.529	0.672
0.84					0.000	0.026	0.053	0.079	0.106	0.134	0.162	0.190	0.220	0.251	0.283	0.317	0.354	0.395	0.443	0.503	0.646
0.85						0.000	0.027	0.053	0.080	0.108	0.136	0.164	0.194	0.225	0.257	0.291	0.328	0.369	0.417	0.477	0.620
0.86							0.000	0.026	0.053	0.081	0.109	0.137	0.167	0.198	0.230	0.264	0.301	0.342	0.390	0.450	0.593
0.87								0.000	0.027	0.055	0.083	0.111	0.141	0.172	0.204	0.238	0.275	0.316	0.364	0.424	0.567
0.88									0.000	0.028	0.056	0.084	0.114	0.145	0.177	0.211	0.248	0.289	0.337	0.397	0.540
0.89										0.000	0.028	0.056	0.086	0.117	0.149	0.183	0.220	0.261	0.309	0.369	0.512
0.90											0.000	0.028	0.058	0.089	0.121	0.155	0.192	0.233	0.281	0.341	0.484
0.91												0.000	0.030	0.061	0.093	0.127	0.164	0.205	0.253	0.313	0.456
0.92													0.000	0.031	0.063	0.097	0.134	0.175	0.223	0.283	0.426
0.93														0.000	0.032	0.066	0.103	0.144	0.192	0.252	0.395
0.94															0.000	0.034	0.071	0.112	0.160	0.220	0.363
0.95																0.000	0.037	0.079	0.026	0.186	0.329
0.96																	0.000	0.041	0.089	0.149	0.292
0.97																		0.000	0.048	0.108	0.251
0.98																			0.000	0.060	0.203
0.99																				0.000	0.143
																					0.000

Location of Capacitors

The primary purpose of capacitors is to reduce the maximum demand. Additional benefits are derived by capacitor location. The Figure 1.13 indicates typical capacitor locations. Maximum benefit of capacitors is derived by locating them as close as possible to the load. At this location, its kilovars are confined to the smallest possible segment, decreasing the load current. This, in turn, will reduce power losses of the system substantially. Power losses are proportional to the square of the current. When power losses are reduced, voltage at the motor increases; thus, motor performance also increases.

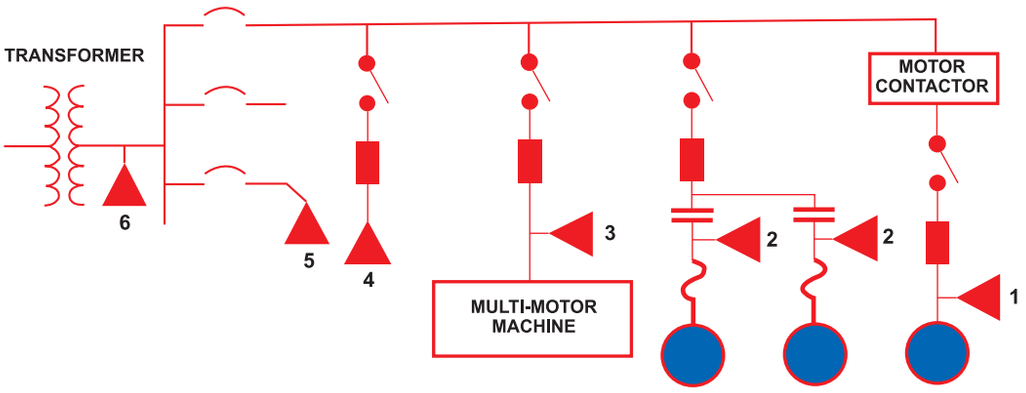


Figure 1.13 Power Distribution Diagram Illustrating Capacitor Locations

Where to correct power factor?

Capacitor correction is relatively inexpensive both in material and installation costs. Capacitors can be installed at any point in the electrical system, and will improve the power factor between the point of application and the power source. However, the power factor between the utilization equipment and the capacitor will remain unchanged. Capacitors are usually added at each piece of offending equipment, ahead of groups of small motors (ahead of motor control centers or distribution panels) or at main services.

The advantages and disadvantages of each type of capacitor installation are listed below:

Capacitor on each piece of equipment (1,2)

Advantages

- Increases load capabilities of distribution system.
- Can be switched with equipment; no additional switching is required.
- Better voltage regulation because capacitor use follows load.
- Capacitor sizing is simplified.
- Capacitors are coupled with equipment and move with equipment if rearrangements are instituted.

Disadvantages

- Small capacitors cost more per kVAr than larger units (economic break point for individual correction is generally at 10 HP).

It should be noted that the rating of the capacitor should not be greater than the no-load magnetizing kVAr of the motor. If this condition exists, damaging over voltage or transient torques can occur. This is why most motor manufacturers specify maximum capacitor ratings to be applied to specific motors.

Capacitor with equipment group (3)

Advantages

- Increased load capabilities of the service
- Reduced material costs relative to individual correction
- Reduced installation costs relative to individual correction

Disadvantages

- Switching means may be required to control amount of capacitance used.

The advantage of locating capacitors at power centers or feeders is that they can be grouped together. When several motors are running intermittently, the capacitors are permitted to be on line all the time, reducing the kVA demand regardless of load.

Capacitor at main service (4,5, & 6)

Advantages

- Low material installation costs.

Disadvantages

- Switching will usually be required to control the amount of capacitance used.
- Does not improve the load capabilities of the distribution system.

From energy efficiency point of view, capacitor location at receiving substation only helps the utility in loss reduction. Locating capacitors at tail end will help to reduce loss reduction within the plants distribution network as well and directly benefit the user by reduced consumption. Reduction in the distribution loss% in kWh when tail end power factor is raised from PF_1 to a new power factor PF_2, will be proportional to

$$\left[1-(PF_1/PF_2)^2\right] \times 100$$

Other Considerations

Where the loads contributing to power factor are relatively constant, and system load capabilities are not a factor, correcting at the main service could provide a cost advantage. When the low power factor is derived from a few selected pieces of equipment, individual equipment correction would be cost effective.

The growing use of ASDs (nonlinear loads) has increased the complexity of system power factor and its corrections. The application of PF correction capacitors without a thorough analysis of the system can aggravate rather than correct the problem, particularly if the fifth and seventh harmonics are present.

Capacitors for Other Loads

The other types of load requiring capacitor application include induction furnaces, induction heaters and arc welding transformers etc. The capacitors are normally supplied with control gear for the application of induction furnaces and induction heating furnaces. The PF of arc furnaces experiences a wide variation over melting cycle as it changes from 0.7 at starting to 0.9 at the end of the cycle. Power factor for welding transformers is corrected by connecting capacitors across the primary winding of the transformers, as the normal PF would be in the range of 0.35.

Performance Assessment of Power Factor Capacitors

Voltage effects: Ideally capacitor voltage rating is to match the supply voltage. If the supply voltage is lower, the reactive kVAr produced will be the ratio V_1^2 / V_2^2 where V_1 is the actual supply voltage, V_2 is the rated voltage.

On the other hand, if the supply voltage exceeds rated voltage, the life of the capacitor is adversely affected.

Material of capacitors: Power factor capacitors are available in various types by dielectric material used as; paper/ polypropylene etc. The watt loss per kVAr as well as life vary with respect to the choice of the dielectric material and hence is a factor to be considered while selection.

Connections: Shunt capacitor connections are adopted for almost all industry/ end user applications, while series capacitors are adopted for voltage boosting in distribution networks.

Operational performance of capacitors: This can be made by monitoring capacitor charging current vis- a- vis the rated charging current. Capacity of fused elements can be replenished as per requirements. Portable analyzers can be used for measuring kVAr delivered as well as charging current. Capacitors consume 0.2 to 6.0 Watt per kVAr, which is negligible in comparison to benefits.

Some checks that need to be adopted in use of capacitors are:

i. Nameplates can be misleading with respect to ratings. It is good to check by charging currents.

ii. Capacitor boxes may contain only insulated compound and insulated terminals with no capacitor elements inside.

iii. Capacitors for single phase motor starting and those used for lighting circuits for voltage boost, are not power factor capacitor units and these cannot withstand power system conditions.

1.5 Transformers

A transformer can accept energy at one voltage and deliver it at another voltage. This permits electrical energy to be generated at relatively low voltages and transmitted at high voltages and low currents, thus reducing line losses and voltage drop (see Figure 1.14).

Figure 1.14 View of a Transformer

Transformers consist of two or more coils that are electrically insulated, but magnetically linked. The primary coil is connected to the power source and the secondary coil connects to the load. The turn's ratio is the ratio between the numbers of turns on the secondary to the turns on the primary (See Figure 1.15).

The secondary voltage is equal to the primary voltage times the turn's ratio. Ampere-turns are calculated by multiplying the current in the coil times the number of turns. Primary ampere-turns are equal to secondary ampere-turns. Voltage regulation of a transformer is the percent increase in voltage from full load to no load.

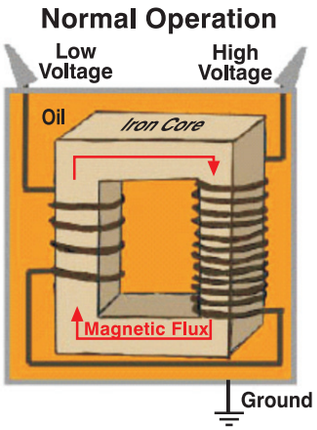


Figure 1.15 View of a Transformer Coil

Types of Transformers

Transformers are classified as two categories: power transformers and distribution transformers.

Power transformers are used in transmission network of higher voltages, deployed for step-up and step down transformer application (400 kV, 200 kV, 110 kV, 66 kV, 33kV)

Distribution transformers are used for lower voltage distribution networks as a means to end user connectivity. (11kV, 6.6 kV, 3.3 kV, 440V, 230V)

Rating of Transformer

Rating of the transformer is calculated based on the connected load and applying the diversity factor on the connected load, applicable to the particular industry and arrive at the kVA rating of the Transformer. Diversity factor is defined as the ratio of overall maximum demand of the plant to the sum of individual maximum demand of various equipment. Diversity factor varies from industry to industry and depends on various factors such as individual loads, load factor and future expansion needs of the plant. Diversity factor will always be less than one.

Location of Transformer

Location of the transformer is very important as far as distribution loss is concerned. Transformer receives HT voltage from the grid and steps it down to the required voltage. Transformers should be placed close to the load centre, considering other features like optimization needs for centralized control, operational flexibility etc. This will bring down the distribution loss in cables.

Transformer Losses and Efficiency

The efficiency varies anywhere between 96 to 99 percent. The efficiency of the transformers not only depends on the design, but also, on the effective operating load.

Transformer losses consist of two parts: No-load loss and Load loss

1. No-load loss (also called core loss) is the power consumed to sustain the magnetic field in the transformer's steel core. Core loss occurs whenever the transformer is energized; core loss does not vary with load. Core losses are caused by two factors: hysteresis and eddy current losses. Hysteresis loss is that energy lost by reversing the magnetic field in the core as the magnetizing AC rises and falls and reverses direction. Eddy current loss is a result of induced currents circulating in the core.
2. Load loss (also called copper loss) is associated with full-load current flow in the transformer windings. Copper loss is power lost in the primary and secondary windings of a transformer due to the ohmic resistance of the windings. Copper loss varies with the square of the load current. ($P=I^2R$). Typical 3 Phase Transformer losses of various capacities is given in Table1.3.

Transformer losses as a percentage of load is given in the Figure 1.16.

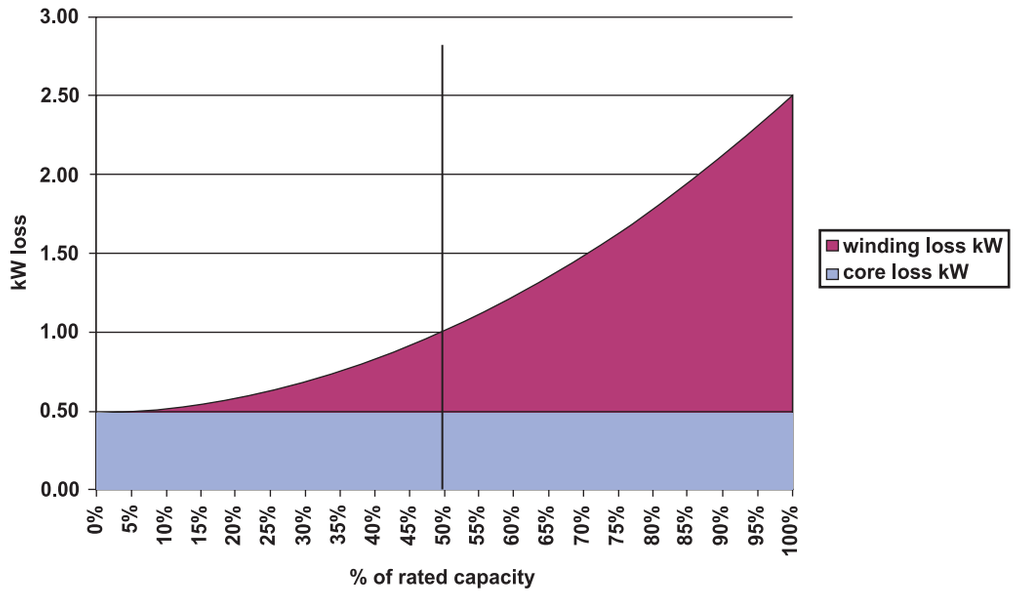


Figure 1.16 Transformer loss vs %Load

For a given transformer, the manufacturer can supply values for no-load loss, $P_{NO\text{-}LOAD}$, and load loss, P_{LOAD}. The total transformer loss, P_{TOTAL}, at any load level can then be calculated from:

$$\mathbf{P_{TOTAL} = P_{NO\text{-}LOAD} + (\%\ Load/100)^2 \times P_{LOAD}}$$

Where transformer loading is known, the actual transformers loss at given load can be computed as:

$$= \text{No load loss} + \left(\frac{\text{kVA load}}{\text{Rated kVA}}\right)^2 \text{x (Full load loss)}$$

Table 1.3 – Typical 3 Phase Transformer losses of various capacities (for CRGO Core Transformers)		
Rating (KVA)	**No Load Loss (W)**	**Load Loss (W)**
100	320	1950
160	455	2800
250	640	4450
500	900	6450
630	1260	9300
1000	1800	13300
1600	2600	19800
2000	3200	21000
3150	4600	28000
5000	6500	38000
6300	7700	45000
10000	11000	63000
12500	13000	77000
20000	18000	107000
31500	25000	150000
40000	30000	180000

Source: Siemens Electrical Engineers Hand Book

Voltage Fluctuation Control

A control of voltage in a transformer is important due to frequent changes in supply voltage level. Whenever the supply voltage is less than the optimal value, there is a chance of nuisance tripping of voltage sensitive devices. The voltage regulation in transformers is done by altering the voltage transformation ratio with the help of tapping.

There are two methods of tap changing facility available: ***Off-circuit tap changer and On-load tap changer.***

Off-circuit tap changer

It is a device fitted in the transformer, which is used to vary the voltage transformation ratio. Here the voltage levels can be varied only after isolating the primary voltage of the transformer.

On load tap changer (OLTC)

The voltage levels can be varied without isolating the connected load to the transformer. To minimize the magnetization losses and to reduce the nuisance tripping of the plant, the main transformer (the transformer that receives supply from the grid) should be provided with On Load Tap Changing facility at design stage. The downstream distribution transformers can be provided with off-circuit tap changer.

The On-load gear can be put in auto mode or manually depending on the requirement. OLTC can be arranged for transformers of size 250 kVA onwards. However, the necessity of OLTC below 1000 kVA can be considered after calculating the cost economics.

Parallel Operation of Transformers

The design of Power Control Centre (PCC) and Motor Control Centre (MCC) of any new plant should have the provision of operating two or more transformers in parallel. Additional switchgears and bus couplers should be provided at design stage.

Whenever two transformers are operating in parallel, both should be technically identical in all aspects and more importantly should have the same impedance level. This will minimize the circulating current between transformers.

Where the load is fluctuating in nature, it is preferable to have more than one transformer running in parallel, so that the load can be optimized by sharing the load between transformers. The transformers can be operated close to the maximum efficiency range by this operation.

For operating transformers in parallel, the transformers should have the following principal characteristics.

- The same phase angle difference between the primary and secondary terminals.
- Same voltage ratio
- Same percentage impedance
- Same polarity
- Same phase sequence

Energy Efficient Transformers

Most energy loss in dry-type transformers occurs through heat or vibration from the core. The new high-efficiency transformers minimize these losses. The conventional transformer is made up of a silicon alloyed iron (grain oriented) core. The iron loss of any transformer depends on the type of core used in the transformer. However the latest technology is to use amorphous material – a metallic glass alloy for the core (see Figure 1.17). The expected reduction in core loss over conventional (Si Fe core) transformers is roughly around 70%, which is quite significant. By using an amorphous core– with unique physical and magnetic properties- these new types of transformers have increased efficiency even at low loads - 98.5% efficiency at 35% load.

Electrical distribution transformers made with amorphous metal cores provide excellent opportunity to conserve energy right from the installation. Though these transformers are a little costlier than conventional iron core transformers, the overall benefit towards energy savings will compensate for the higher initial investment.

Figure 1.17 1600 kVA Amorphous Core Transformer

Standards & Labeling Programme for Distribution Transformers

The Bureau of Energy Efficiency has included Distribution transformers under Standards & Labeling Programme as large number of Distribution transformers are used by Electricity supply companies and also by different users for supplying power to their load centers. This provision has been made mandatory with effect from 7th January 2010.

The existing efficiency or the loss standards are specified in IS 1180 (part 1). This standard defines load losses and no load losses separately. For the BEE labeling programme total losses at 50% and 100% load have been defined. The highest loss segment is defined as star 1 and lowest loss segment is defined as star 5. The existing IS 1180 (part 1) specification losses are the base case with star 1.

The details of Star Rating plan for Distribution transformers and corresponding losses are given in Table 1.4. More details can be obtained from www.beestarlabel.com.

Table 1.4 Total Transformer Losses at 50% and 100% Loading

	1 star		2 star		3 star		4 star		5 star	
Rating kVA	Max Losses at 50% (Watts)	Max Losses at 100% (Watts)	Max Losses at 50% (Watts)	Max Losses at 100% (Watts)	Max Losses at 50% (Watts)	Max Losses at 100% (Watts)	Max Losses at 50% (Watts)	Max Losses at 100% (Watts)	Max Losses at 50% (Watts)	Max Losses at 100% (Watts)
16	200	555	165	520	150	480	135	440	120	400
25	190	785	235	740	210	695	190	635	175	595
63	490	1415	430	1335	380	1250	340	1140	300	1050
100	700	2020	610	1910	520	1800	475	1650	435	1500
160	1000	2800	880	2550	770	2200	670	1950	570	1700
200	1130	3300	1010	3000	890	2700	780	2300	670	2100

1.6 Distribution Losses in Industrial System

In an electrical system often the constant no load losses and the variable load losses are to be assessed, over long reference duration, for energy loss estimation.

Identifying and calculating the sum of the individual contributing loss components is a challenging one, requiring extensive experience and knowledge of all the factors impacting the operating efficiencies of each of these components.

For example the cable losses in any industrial plant will be up to 6 percent depending on the size and complexity of the distribution system. All of these are current dependent, and can be readily mitigated by any technique that reduces facility current load. The various losses in different distribution equipments are given in Table1.5.

In system distribution loss optimization, the various options available include:

- Relocating transformers and sub-stations near to load centers
- Re-routing and re-conducting such feeders and lines where the losses / voltage drops are higher.

- Power factor improvement by incorporating capacitors at load end.
- Optimum loading of transformers in the system.
- Opting for lower resistance All Aluminum Alloy Conductors (AAAC) in place of conventional Aluminum Cored Steel Reinforced (ACSR) lines
- Minimizing losses due to weak links in distribution network such as jumpers, loose contacts, and old brittle conductors.

Table 1.5 Losses in Electrical Distribution Equipment

S.No	Equipment	% Energy Loss at Full Load Variations	
		Min	Max
1.	Outdoor circuit breaker (15 to 230 KV)	0.002	0.015
2.	Generators	0.019	3.5
3.	Medium voltage switchgears (5 to 15 KV)	0.005	0.02
4.	Current limiting reactors	0.09	0.30
5.	Transformers	0.40	1.90
6.	Load break switches	0.003	0.025
7.	Medium voltage starters	0.02	0.15
8.	Bus ways less than 430 V	0.05	0.50
9.	Low voltage switchgear	0.13	0.34
10.	Motor control centers	0.01	0.40
11.	Cables	1.00	4.00
12.	Large rectifiers	3.0	9.0
13.	Static variable speed drives	6.0	15.0
14.	Capacitors (Watts / kVAr)	0.50	6.0

1.7 Assessment of Transmission and Distribution (T&D) Losses in Power Systems

For an electric utility (DISCOMs) the distribution losses which are more predominant, can be categorized as

i) Technical Losses

ii) Commercial Losses

Technical Losses:

The technical losses primarily take place due to the following factors

- Transformation Losses (at various transformation levels)
- High I^2R losses in distribution lines due to inherent resistance and poor power factor in the electrical network

Normative Technical loss limits in Indian Transmission and Distribution network are shown in Table 1.6.

Table 1.6 Normative Technical loss limits in Transmission and Distribution network in Indian Context

System Component	Loss Limit % Min	Loss Limit % Max
STEP-UP Transformers & EHV Transmission System	0.5	1.0
Transmission to intermediate voltage level, transmission system & Step-down to sub transmission Voltage level	1.5	3.0
Sub transmission System & step down to distribution voltage level	2.0	4.5
Distribution lines and Service connections	3.0	7.0
Total Losses	**7.0**	**15.5**

1.8 Estimation of Technical Losses in Distribution System

The first and important step in reduction of energy losses is to carry out energy audit of power distribution system. There are two methods of determining the energy losses namely direct method and indirect method.

The **Direct method** involves placement of energy meters at all locations starting from the input point of the feeder to the individual consumers. The difference between input energy and sum of all consumers over a specific duration is accounted as distribution loss of the network. This calls for elaborate and accurate metering and collection of simultaneous data.

The **Indirect method** essentially involves:

- Energy metering at critical locations in the system such as substation and feeders.
- Compiling the network information, such as length of the line/feeders, conductor size, DTR details, capacitor details etc.
- Conducting load flow studies (all electrical parameters) on peak load durations as well as normal load durations.
- Application of suitable software to assess the system losses.

This software can also be used for system simulation, identifying improvements and network optimization.

Causes of technical losses in distribution system

The factors contributing to the increase in the distribution losses are

1. Lengthy distribution lines:

In practice, 11 KV and 415 volts lines, in rural areas are extended radially over long distances to feed loads scattered over large areas. This results in high line resistance and therefore high I^2R losses in the line.

2. Inadequate Size of Conductors:

On account of load growth, many distribution feeders end up being under sized for the loads to be catered to the consumers. The size of the conductors should be selected/upgraded/transformers to be relocated on the basis of KVA Kilometer capacity of standard conductor to maintain voltage regulation within limits.

Voltage Regulation:

The voltage regulation is usually expressed as a percentage drop with reference to the receiving end voltage.

Percentage regulation = 100 (Es - Er) / Er

Where, Es = Sending end voltage
Er = Receiving end voltage

3. Distribution Transformers (DTR) not located at load center on the Secondary Distribution System:

Often, DTs are not located centrally with respect to consumer loads. Consequently, the farthest consumers receive low voltage even though a good voltage level is maintained at the transformer's secondary. This again leads to high line loss. Therefore in order to reduce the voltage drop in the line to the farthest consumers, the distribution transformer should be located near to consumer load to keep voltage drop within permissible limits.

4. Low Power Factor:

A low PF contributes towards high distribution losses. For a given load, if the PF is low, the current drawn is high. Consequently, the losses which are proportional to square of the current will be more.

Therefore, line losses owing to the poor PF can be reduced by improving the PF. This can be done by application of shunt capacitors.

Shunt capacitors can be connected in the following locations:

- On the secondary side (11 KV side) of the 33/11 KV power transformers in substation.
- On the secondary side of distribution transformers

The following example shows how the improvement in power factor in 11 KV lines results in considerable reduction in losses:

Reduction of Line Losses with improvement in Power Factor

Load (kW)	PF	kVA	Current (A)	Line Loss (kW)	Remarks
300	0.7	428	38.9	27.2	Before
300	1.0	300	27.2	13.4	After

Measures to reduce technical losses

Some of the measures to reduce technical losses in distribution system include,

- **High Voltage Distribution System (HVDS):-** Distribution Companies (Discoms) have started implementing distribution systems at high voltage. The L.T. distributions are reduced and eliminated wherever feasible. A typical LT System consists of LT 3 Phase 415V Distribution System with lengthy LT Lines serving the consumers, contributing to more losses in the System. Reduction in these losses is done through restructuring of the existing LVDS network to HVDS network by installation of three phase 11 kV/400V 25 KVA & 16KVA pole mounted transformers at the load centers to serve different consumers.

- **Amorphous Core Transformers:** Recently Distribution Transformers DTRs with amorphous core have been manufactured with just about 30% of no-load losses compared to the Conventional Transformers. Some of the Discoms have installed these transformers to reduce the distribution loess in the network.

Commercial Losses

Any illegal consumption of electrical energy, which is not correctly metered, billed and revenue collected, causes commercial losses to the utilities. The commercial losses are primarily attributable to discrepancies in:

Meter Reading: Commercial losses occur due to discrepancy in meter reading. Meter reading problems are manifested in the form of zero consumption in meter reading books which may be due to premises found locked, untraceable consumers, stopped/defective meters, temporarily disconnected consumers continuing in billing solution etc. Collusion with consumers is also a source of commercial loss to utilities which are primarily due to incorrect meter reading.

Metering: Most utilities use either electro-mechanical or electronic meters for consumer metering. Commercial losses through metering can be in the form of meter tampering in various forms.

Collection efficiency: Typically in a billing cycle, a distribution utility issues bills against metered energy and assessed (generally in case of agricultural loads and temporary connections) energy. The ratio of amount collected to total amount billed is termed as collection efficiency.

The above losses are collectively categorized as AT & C (Aggregate Technical & Commercial) losses. The estimation of AT & C losses for a sample area is shown in Table 1.7.

Computation of AT & C Losses

The aggregate technical and commercial losses can be measured using the formula mentioned below.

$$\text{AT \& C Losses} = \{1 - (\text{Billing Efficiency} \times \text{Collection Efficiency})\} \times 100$$

Where,

$$\text{Billing Efficiency, \%} = \frac{\text{Total units sold, } MU}{\text{Total input, } MU} \times 100$$

$$\text{Collection Efficiency, \%} = \frac{\text{Re } venue \text{ } Collected, \text{ } Rs.}{Amount \text{ } billed, \text{ } Rs.} \times 100$$

MU = Million Units

Table 1.7 Estimation of AT & C Losses			
Sl. No.	**Description**		**Annual data**
1	Input Energy = (Import-Export), MU	Ei	10
2a	Energy Billed (Metered), MU	E1	6
2b	Energy Billed (Un-Metered), MU	E2	1
2c	Total Energy Billed (E1 + E2)	Eb	7
3	Amount Billed (Rs. lakhs)	Ab	400
4a	Gross Amount Collected (Rs. lakhs)	AG	410
4b	Arrears Collected (Rs. lakhs)	Ar	40
4c	Amount Collected without Arrears (Rs. lakhs)	Ac=AG-Ar	370
5	Billing Efficiency (BE)	= Eb/Ei *100%	70%
6	Collection Efficiency(CE)	=Ac/Ab *100%	93%
7	AT& C Loss	{1- (BE *CE) *100%	35%

BE = Billing Efficiency, CE = Collection Efficiency, Note: If A_r is not known, assume $A_r = A_G - A_b$

Measures to Reduce Commercial Losses

Some of the measures to reduce commercial losses in distribution system include:

- Accurate Metering (A metering plan for installing meters with sustained accuracy).
- Appropriate range of meter with reference to connected load.
- Installation of Electronic meters with (TOD, tamper proof, data and remote reading facility).
- Intensive inspections.
- Compulsory metering/average billing
- Use of energy audit as a tool to pinpoint areas of high losses.
- Eradication of theft.

1.9 Demand Side Management (DSM)

DSM refers to "Actions taken on the customer's side of the meter to change the amount (kWh) or timing (kVA) of energy consumption. Electricity DSM strategies have the goal of maximizing end use efficiency to avoid or postpone the construction of new generating plants".

The ever increasing demand growth of electricity can be met either by matching increase in capacity, i.e. Supply side capacity addition or adopting demand side management and end use efficiency improvement strategies, which are much more cost effective and resource efficient.

Utilities are driven by supply side and customer side concerns such as capacity (peak demand) short falls, energy shortfalls, need for optimization of generation and network utilization, Regulatory issues, environmental mandates and customer demand for uninterrupted supply at competitive tariffs. Demand side management offers itself as a powerful tool to distribution companies, to analyze, develop and implement customized DSM programs, cost effectively, to enable meeting the supply side concern of the utilities.

DSM Objectives

The key objectives of DSM include the following.

- Improve the efficiency of energy systems.
- Reduce financial needs to build new energy facilities (generation).
- Minimize adverse environmental impacts.
- Lower the cost of delivered energy to consumers.
- Reduce power shortages and power cuts.
- Improve the reliability and quality of power supply.

DSM methodology

Step 1: Load Research

This stage in the DSM implementation will typically assess the customer base, tariff, load profile on an hourly basis and will identify the sectors contributing to the load shape. This step will also identify peak load contributors.

Step 2: Define load-shape objectives

Based on the results of the load research in the utility, the load shape objectives for the current situation are defined. Various load-shape objectives are represented in Figure 1.18 below.

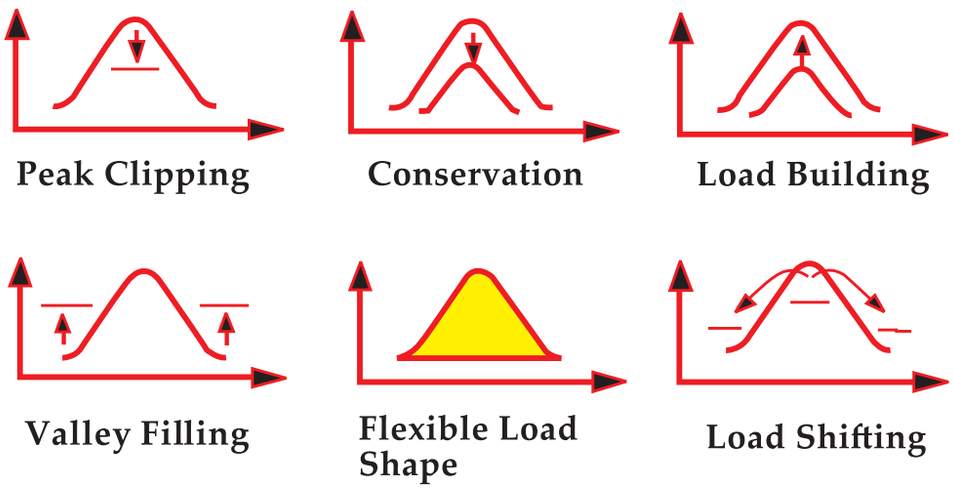


Figure 1.18 Load Shape Profiles and Objectives

Meaning of load-shape objective

Peak Clipping: the reduction of utility load primarily during periods of peak demand.

Valley-Filling: the improvement of system load factor by building load in off-peak periods.

Load Shifting: the reduction of utility loads during periods of peak demand, while at the same time building load in off-peak periods. Load shifting typically does not substantially alter total electricity sales.

Conservation: the reduction of utility loads, more or less equally, during all or most hours of the day.

Load Building: the increase of utility loads, more or less equally, during all or most hours of the day.

Flexible Utility Load Shape: refers to programs that set up utility options to alter customer energy consumption on an as-needed basis, as in interruptible/ curtailable agreements.

Step 3: Assess program implementation strategies

This step will identify the end-use applications that can be potentially targeted to reduce peak demand, specifically in sectors contributing to system peak. This step will also carry out a detailed cost *benefit analysis* for the end-users and the utilities, including analysis of societal as well as environmental benefits.

Step 4: Implementation

Implementation stage includes program design for specific end-use applications, promotes the program to the target audience through marketing approaches such as advertising, bills and inserts, and focused group meetings (specifically in case of commercial and industrial sector).

Step 5: Monitoring and Evaluation

This step tracks the program design and implementation and compares the same with proposed DSM goal set by the utility. A detailed benefit-cost analysis in this case includes identifying the avoided supply cost for the utility vis-à-vis the total program cost for the utilities and benefits to the participants including the reduced bills or incentives to the end-users.

Types of DSM Measures

Broadly the types of DSM measures can be classified as follows.

a) Energy reduction programmes - reducing demand through more efficient processes, buildings or equipment, for example:

Efficient Lighting (CFLs, Using natural light), Appliance Labelling, Building regulations, Efficient and alternative energy use, Efficient use of electric motors and motor driven systems, Preventative maintenance, Energy management and audit.

b) Load management programmes - changing the load pattern and encouraging less demand at peak times and peak rates, for example:

Load Levelling (Peak clipping, Valley filling and load shifting), Load growth, Tariff Incentives or Penalties (Time-of-Use & real time pricing, power factor penalties)

Benefits of DSM

Benefits of the DSM initiatives are manifold, some of which are described in Table 1.8 below.

Table 1.8 Benefits of DSM

Customer benefits	Societal benefits	Utility benefits
Satisfy electricity demands Reduce / stabilize costs (bills) Improve value of service Maintain/improve lifestyle and productivity	Reduce environmental degradation Conserve resources Protect global environment Maximize customer welfare	Lower cost of service Improve operating efficiency, flexibility Reduce capital needs Improve customer service

1.10 Harmonics

A harmonic is a component frequency of the signal that is an integer multiple of the fundamental frequency. Harmonic voltages and currents in an electric power system are a result of non-linear electric loads. **The Harmonic current represents energy that cannot be used by any devices on the network. It will be therefore converted to heat and is wasted.** For instance, the fundamental frequency is 50 Hz, and then the 5th harmonic is five times that frequency, or 250 Hz (Figure 1.19).

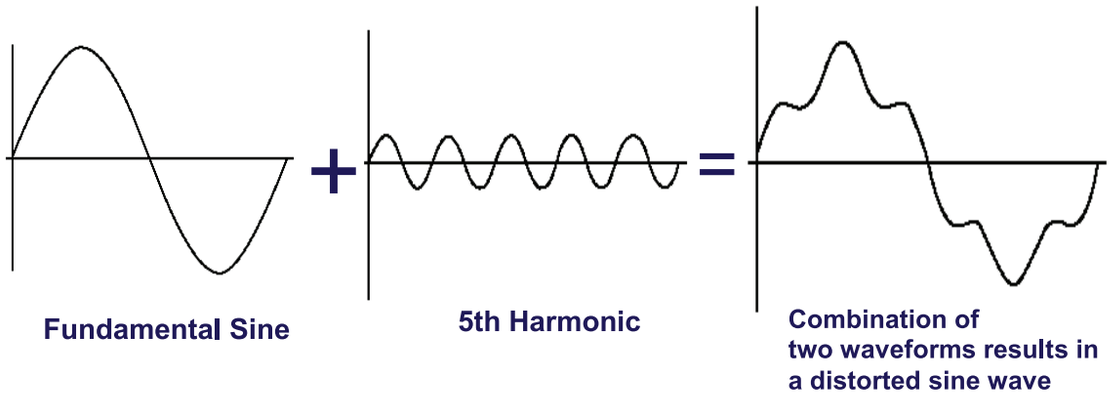


Figure 1.19 Harmonic wave pattern

Linear System

In any alternating current network, flow of current depends upon the voltage applied and the impedance (resistance to AC) provided by elements like resistances, reactances of inductive and capacitive nature. As the value of impedance in above devices is constant, they are called linear whereby the voltage and current relation is of linear nature.

e.g. Incandescent lamps, heaters and, to a great extent, motors are linear systems

Non-linear System

Non-Linear systems are one with varying impedance characteristics, These NON LINEAR devices cause distortion in voltage and current waveforms which is of increasing concern in recent times.

e.g. Variable frequency drives (VFDs), electronic ballasts, UPS and Computers, induction and arc furnaces

Current Distortion:

Current Harmonics could cause transformer heating or nuisance tripping by fuses, circuit breakers and other protective devices since they are typically not rated for harmonically rich waveforms.

$$THD_{current} = \sqrt{\sum_{n=2}^{n=n}\left(\frac{I_n}{I_1}\right)^2} \times 100$$

A 5th harmonic current is simply a current flowing at 250 Hz on a 50 Hz system. The 5th harmonic current flowing through the system impedance creates a 5th harmonic voltage. Total Harmonic Distortion (THD) expresses the amount of harmonics. The following is the formula for calculating the THD for current:

$$THD_{current} = \sqrt{\sum_{n=2}^{n=n}\left(\frac{I_n}{I_1}\right)^2} \times 100$$

Then…

$$I_{THD} = \sqrt{\left[\left(\frac{50}{250}\right)^2 + \left(\frac{35}{250}\right)^2\right]} x100 \;=\; 24\%$$

Current at fundamental frequency I_1 = Base Current = 250 amps
Third Harmonic current = 50 amps
Fifth Harmonic current = 35 amps

When harmonic currents flow in a power system, they are known as “poor power quality” or “dirty power”. Other causes of poor power quality include transients such as voltage spikes, surges, sags, and ringing. Because they repeat every cycle, harmonics are regarded as a steady-state cause of poor power quality. The distortion travels back into the power source and can affect other equipment connected to the same source.

Voltage Distortion:

A distorted current has higher peak values that cause non-sinusoidal voltage drops across the distribution system. The resulting voltage drops add or subtract from the sinusoidal voltage supplied by the utility. Other utility customers could get distorted voltage on the downstream side of the power distribution circuit.

When expressed as a percentage of fundamental voltage THD is given by,

$$THD_{voltage} = \sqrt{\sum_{n=2}^{n=n}\left(\frac{V_n}{V_1}\right)^2} \times 100$$

Where, V_1 is the fundamental frequency voltage and V_n is n^{th} harmonic voltage component.

Causes of Harmonics

Harmonic currents and voltages are created by non- linear loads connected to the power distribution system. All power electronic converters used in different types of electronic systems can increase harmonic disturbances by injecting harmonic currents directly into the supply network. Common non-linear loads include variable speed drives (AC as well as DC), induction furnaces, LED based and CFL lamps, certain types of UPS & computer power supplies.

Effects of Harmonics

- Blinking of Incandescent Lights - Transformer Saturation
- Capacitor Failure - Harmonic Resonance
- Circuit Breakers Tripping - Inductive Heating and Overload
- Conductor Failure - Inductive Heating
- Electronic Equipment Shutting down - Voltage Distortion
- Flickering of Fluorescent Lights - Transformer Saturation
- Fuses Blowing for No Apparent Reason - Inductive Heating and Overload
- Motor Failures (overheating) - Voltage Drop
- Neutral Conductor and Terminal Failures - Additive Triplen Currents
- Electromagnetic Load Failures - Inductive Heating
- Overheating of Metal Enclosures - Inductive Heating
- Power Interference on Voice Communication - Harmonic Noise
- Transformer Failures - Inductive Heating

Overcoming Harmonics in Power systems

Passive Filters: Built-up by combinations of capacitors, inductors (reactors) and resistors. It is the most common and available for all voltage levels

Active Power Filter APF: Inserting negative phase compensating harmonics into the AC-Network, thus eliminating the undesirable harmonics on the AC power network.

Special Transformers: There are several special types of transformer connections which can cancel harmonics. Additional special winding connections can be used to cancel other harmonics on balanced loads. These systems also use more copper. Harmonic canceling transformers are also known as phase-shifting transformers. It is a relatively new power quality product for mitigating harmonic problems.

Since VFD is emerging as a major energy saving application and results in harmonic generation, harmonic mitigation in VFD is discussed here.

The Harmonic Mitigation solutions currently in use in the industry broadly fall into the following categories:

1. Passive Harmonic Filter (PHF)
2. Advance Active Filters (AAF)
3. Active Front End based VFDs (AFE)

They are briefly described in the sections below:

Passive filter is the most common method for the cancellation of harmonic current in the distributed system. These filters are basically designed on principle either single tuned/double tuned or band pass filter technology. Passive filters (Figure 1.20) offer very low impedance in the network at the tuned frequency to divert all the harmonic current at the tuned frequency.

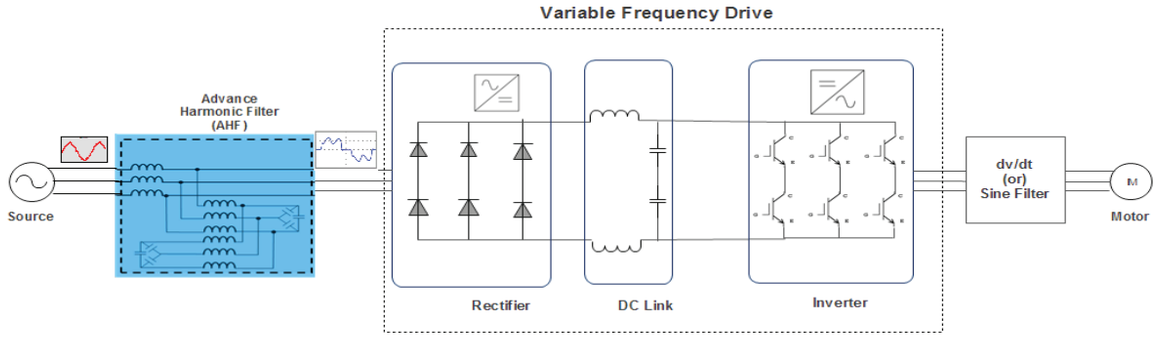


Figure 1.20 Power schematic of Series passive filter

Active filter is connected parallel with the distribution system. Distribution system consists of a wide percentage of harmonics produced by non- linear loads. Active filters (Figure 1.21) compensate current harmonics by injecting equal magnitude but opposite phase harmonic compensating current.

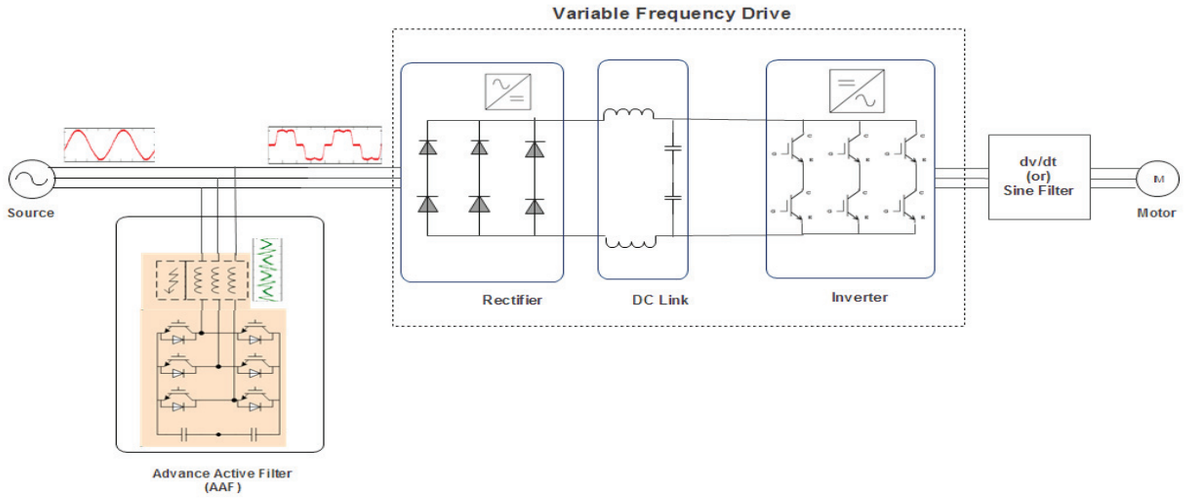


Figure 1.21 Power schematic of shunt active filter

Active Front end Rectifiers used in VFDs has the major advantage of mitigation of harmonics without using external filter, to maintain unity power factor at the point of common coupling, Bidirectional power flow makes recovery of energy to the mains by saving it, Clean power to the grid which in turn does not affect the other loads connected to it, maintaining the DC voltage irrespective of the supply variations.

LCL Filter is connected at Point of Common Coupling (PCC) between Grid and active Front End Rectifier (AFE) (Figure 1.22). Due to High switching frequency operation of AFE IGBTs there will be harmonics in the Sinusoidal waveform at that particular High frequency. The LCL (Inductance and Capacitance Combination) is introduced to bypass those Switching frequencies which will in turn have approximately sinusoidal supply at the Grid.

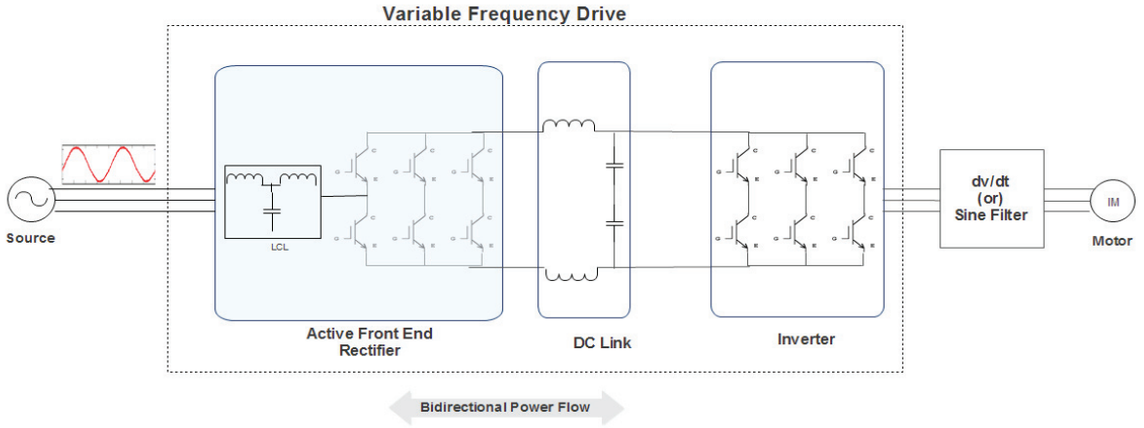


Figure 1.22 Power schematic of Active front end (AFE) drive

Harmonics Limits:

The permissible harmonic limit for different current (Isc / IL) as per IEEE standard is given in Table 1.9 and for different bus voltage are given in Table 1.10

Current Distortion Limits for General Distribution System's end-User limits (120 Volts To 69,000 Volts)

Table 1.9 Maximum Harmonic Current Distortion in % of IL

Individual Harmonic Order (Odd Harmonics)						
Isc/IL	**<11**	**11 ≤ h < 17**	**17 ≤ h < 23**	**23 ≤ h < 35**	**35 ≤ h**	**TDD**
< 20*	4.0	2.0	1.5	0.6	0.3	5.0
20 < 50	7.0	3.5	2.5	1.0	0.5	8.0
50 < 100	10.0	4.5	4.0	1.5	0.7	12.0
100 < 1000	12.0	5.5	5.0	2.0	1.0	15.0
> 1000	15.0	7.0	6.0	2.5	1.4	20.0

Even harmonics are limited to 25% of the odd current harmonic limits above.
Current distortions that result in a direct current offset, e.g. half wave converters are not allowed.
*All power generation equipment is limited to these values of current distortion, regardless of actual Isc/IL.
Where, Isc = Maximum short circuit current at PCC. I_L = Maximum Demand Load Current (fundamental frequency component) at PCC. TDD = Total demand distortion (RSS), harmonic current distortion in % of maximum demand load current (15 or 30 min demand).

Table 1.10 Total Harmonic Distribution for Different Voltage Levels in %

Bus Voltage at PCC	Individual Voltage Distortion (%)	Total Voltage Distortion THD (%)
69 kV and below	3.0	5.0
69.001 kV Thru 161 kV	1.5	2.5
161 kV and above	1.0	1.5

Note:

High voltage systems can have up to 2.0% THD where the cause is an HVDC terminal that will attenuate by the time it is tapped for a user.

Two very important points must be made in reference to the above.

1. The customer is responsible for maintaining current distortion to within acceptable levels, while the utility is responsible for limiting voltage distortion.

2. The limits are only applicable at the point of common coupling (PCC) between the utility and the customer. The PCC, while not explicitly defined, is usually regarded as the point at which the utility equipment ownership meets the customer's or the metering point.

Therefore, the above limits cannot be meaningfully applied to distribution panels or individual equipment within a plant. The entire plant must be considered complying with these limits.

1.11 Analysis of Electrical Power Systems

An analysis of an electrical power system may uncover energy waste, fire hazards, and equipment failure. Facility /energy managers increasingly find that reliability-centered maintenance can save money, energy, and downtime (see Table 1.11).

Table 1.11 Trouble shooting of Electrical Power Systems

System Problem	Common Causes	Possible Effects	Solutions
Voltage imbalances among the three phases	Improper transformer tap settings, Single-phase loads not balanced among phases, poor connections, bad conductors, transformer grounds or faults.	Motor vibration, premature motor failure A 5% imbalance causes a 40% increase in motor losses.	Balance loads among phases.
Voltage deviations from rated voltages (too low or high)	Improper transformer settings, Incorrect selection of motors.	Over-voltages in motors reduce efficiency, power factor and equipment life Increased temperature	Correct transformer settings, motor ratings and motor input voltages
Poor connections in distribution or at connected loads.	Loose bus bar connections, loose cable connections, corroded connections, poor crimps, loose or worn contactors	Produces heat, causes failure at connection site, leads to voltage drops and voltage imbalances	Use Infra Red camera to locate hot-spots and correct.
Undersized conductors.	Facilities expanding beyond original designs, poor power factors	Voltage drop and energy wastage.	Reduce the load by conservation scheduling.
Insulation leakage	Degradation over time due to extreme temperatures, abrasion, moisture, chemicals	May leak to ground or to another phase. Variable energy waste.	Replace conductors, insulators
Low Power Factor	Inductive loads such as motors, transformers, and lighting ballasts Non-linear loads, such as most electronic loads.	Reduces current-carrying capacity of wiring, voltage regulation effectiveness, and equipment life.	Add capacitors to counter reactive loads.
Harmonics (non-sinusoidal voltage and/or current wave forms)	Office-electronics, UPSs, variable frequency drives, high intensity discharge lighting, and electronic and core-coil ballasts.	Over-heating of neutral conductors, motors, transformers, switch gear. Voltage drop, low power factors, reduced capacity.	Take care with equipment selection and isolate sensitive electronics from noisy circuits.

Solved Example:

An energy audit of electricity bills of a process plant was conducted. The plant has a contract demand of 5000 kVA with the power supply company. The average maximum demand of the plant is 3850 kVA/ month at a power factor of 0.95. The maximum demand is billed at the rate of Rs.600/kVA/month. The minimum billable maximum demand is 75 % of the contract demand. An incentive of 0.5 % reduction in energy charges component of electricity bill are provided for every 0.01 increase in power factor over and above 0.95. The average energy charge component of the electricity bill per month for the plant is Rs.18 lakhs.

The plant decides to improve the power factor to unity. Determine the power factor capacitor kVAr required, annual reduction in maximum demand charges and energy charge component. What will be the simple payback period if the cost of power factor capacitors is Rs.900/kVAr.

Ans:

kW drawn	3850 x 0.95 = 3657.5 kW
kVAr required to improve power factor from 0.95 to 1	kW (tan θ_1 – tan θ_2)
	kW (tan ($\cos^{-}\theta_1$) – tan ($\cos^{-}\theta_2$)
	3657.5 (tan ($\cos^{-}0.95$) – tan ($\cos^{-}1$)
	3657.5(0.329 - 0)
	1203 x 900 kVAr
Cost of capacitors @Rs.900/kVAr	Rs.10,82,700
Maximum demand at unity power factor	3657.5/1 = 3657.5 kVA
75 % of contract demand	5000x0.75=3750 kVA
Reduction in Demand charges	3850-3750= 100 kVA, as the plant has to pay MD charges on minimum billable demand of 3750, and not on the improved MD of 3657.5 kVA in this case
	100kVA/month x 12 months x Rs.600 kVA/ month= Rs.7,20,000
Percentage reduction in energy charge from 0.95 to 1 @ 0.5 % for every 0.01 increase	2.5 %
Monthly energy cost component of the bill	Rs.18,00,000
Reduction in energy cost component	18,00,000 x (2.5/100)
	Rs.45,000/month
Annual reduction	Rs.45,000 x 12
	Rs.5,40,000
Savings in electricity bill	Rs.7,20,000+ 5,40,000= 12,60,000
Investment	Rs.10,82,700
Payback period	10,82,700/12,60,000
	0.859 years or 10.31months

QUESTIONS	
Objective Type Questions	
1.	If the distribution voltage is raised from 11 kV to 33 kV, the line loss would be lower by a factor a) 1/9 b) 9 c) 3 d) none
2.	The kVAr rating required for improving the power factor of a load operating at 500 kW and 0.85 power factors to 0.95 is_____. a) 145 kVAr b) 500 kVAr c) 50 kVAr d) 100 kVAr
3.	The rating of the capacitor at motor terminals should not be greater than _____. a) magnetizing kVAr of the motor at full load b) magnetizing kVAr of the motor at no load c) magnetizing kVAr of the motor at half load d) magnetizing kVAr of the motor at 75% load
4.	If voltage applied to a 415 V rated capacitors drops by 10%, its VAR output drops by ____. a) 23% b) 87% c) 19% d) 10%
5.	The sum of individual maximum demand of the plant to the sum of individual maximum demand of various equipments is ___. a) load factor b) diversity Factor c) demand Factor d) maximum demand
6.	The approximate kVA rating required for a DG set with 1000 kW connected load, with diversity factor of 1.5 and 84% loading and 0.8 power factor is a) 500 kVA b) 1000 kVA c) 1500 kVA d) 2000 kVA
7.	Commercial losses in distribution net work is due to a) theft b) average billing c) defective meters d) all the above
8.	The Star rating programme of distribution transformers is based on losses at a) no load b) full load c) at 25 % load d) at 50 % & 100 % load
9.	Demand side Management helps a) to reduce the energy losses b) to reduce system peak demand c) to promote energy efficiency among users. d) all the above
10.	If the reactive power drawn by a particular load is zero, it means the load is operating at a) lagging power factor b) leading power factor c) unity power factor d) none of the above
Short Type Questions	
S-1	During April-2003, the plant has recorded a maximum demand of 600 kVA and average PF is observed to be 0.82 lag, the minimum average PF to be maintained is 0.92 lag as per the independent utility supplier and every one % dip in PF attracts a penalty of Rs 10,000/in each month. a) Calculate the improvement in PF for May-2003 by installing 100kVAr capacitors. b) Calculate penalty to be paid if any during May-2003.

S-2	Which is the best location for capacitor banks for power factor improvement from energy conservation point of view? Why?
S-3	List any five problems that can arise due to harmonics in a system.
S-4	Explain why power is generated at lower voltage and transmitted at higher voltages.
S-5	A trivector-meter installed in mini steel plant with 15 minutes cycle has the following inputs during the maximum demand period: **MD Drawn kVA** — **Duration in Minutes** 1000 — 10 2000 — 5 750 — 10 1500 — 5 What is the maximum demand during the 15 minutes interval?
	Long Type Questions
L-1	a) The maximum demand approved by a utility is 5500 kVA and tariff provides for minimum billing demand of 80% of approved. Review of past 12 months records of bills reveals that the monthly maximum demand recorded is around 4200 kVA. Will there be any benefits in surrendering part of contract demand? If so what is the kVA that you recommend for surrendering? Give the costs savings by surrendering demand, if unit rate for kVA demand is Rs 200. b) A process plant consumes of 12500 kWh per month at 0.9 Power Factor (PF). What is the percentage reduction in distribution losses per month if PF is improved up to 0.96 at load end? Assume existing distribution loss as 4% of the plant energy consumption.
L-2	Explain Demand side management and the various techniques used for DSM. What are the benefits of DSM to customers and utilities distribution companies? Discuss various E.E. schemes used in DSM.

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Chapter 2

Electric Motors

Learning Objectives

In this chapter you will learn about

- Motor types and its characteristics
- Motor selection
- Energy efficient motors and factors affecting motor efficiency
- Factors affecting efficiency during rewinding
- Star operation of motors
- Speed controls
- Soft starters with energy saver
- Standards and labeling programme for energy efficient motors

2. ELECTRIC MOTORS

Types, Squirrel cage and slip ring and their characteristics, Losses in induction motors, motor efficiency, Factors affecting motor performance, Rewinding versus replacement of motor, Energy saving opportunities, Motor history sheet, Star operation of motors, Energy efficient motors, Variable speed drives, Soft starters with energy saver, Electrical characteristics. S & L programme on energy efficient motors.

2.1 Introduction

Motors convert electrical energy into mechanical energy by the interaction between the magnetic fields set up in the stator and rotor windings. Industrial electric motors can be broadly classified as induction motors, direct current motors or synchronous motors. All motor types have the same four operating components: stator (stationary windings), rotor (rotating windings), bearings, and frame (enclosure).

2.2 Motor Types

Induction Motors

An AC induction motor (Figure 2.1) has a fixed outer portion, called the stator and a rotor that spins inside with a carefully engineered air gap between the two.

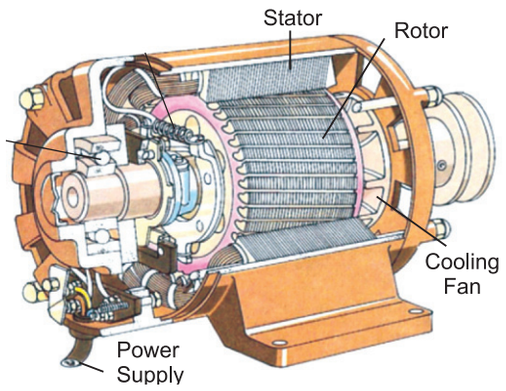

Figure 2.1 Induction Motor

If a 3-phase supply is fed to the stator windings of a 3-phase motor, *a magnetic flux of constant magnitude, rotating at synchronous speed is set up*. At this point, the rotor is stationary. The rotating magnetic flux passes through the air gap between the stator & rotor and sweeps past the stationary rotor conductors. *This rotating flux, as it sweeps, cuts the rotor conductors, thus causing an e.m.f to be induced in the rotor conductors*. As per the *Faraday's law of electromagnetic induction*, it is this relative motion between the rotating magnetic flux and the stationary rotor conductors, which induces an e.m.f on the rotor conductors. Since the rotor conductors are shorted and form a closed circuit, the induced e.m.f produces a rotor current whose direction is given by *Lenz's Law*, is such as to oppose the cause producing it. In this case, the cause which produces the rotor current is the relative motion between the rotating magnetic flux and the stationary rotor conductors. Thus to reduce the relative speed, the rotor starts to rotate in the same direction as that of the rotating flux on the stator windings, trying to catch it up. The frequency of the induced e.m.f is same as the supply frequency.

The magnetic field produced in the rotor because of the induced voltage is alternating in nature. To reduce the relative speed, with respect to the stator, the rotor starts running in the same direction as that of the stator flux and tries to catch up with the rotating flux. However, in practice, the rotor never succeeds in "catching up" to the stator field. The rotor runs slower than the speed of the stator field.

The windings on the rotor of a squirrel cage motor is comprised of aluminum (or sometimes copper) bars embedded in the steel laminations of the rotor. The ends of the rotor bars are shorted together by rings at each end of the rotor. There is no external electrical connection to the rotor. The bar and ring structure looks like an exercise wheel for a pet squirrel.

Slip-ring motor

The slip-ring motor or wound-rotor motor is a variation of the squirrel cage induction motor. While the stator is the same as that of the squirrel cage motor, the rotor of a slip-ring motor is wound with wire coils. The ends of the windings are connected to slip rings so that resistors or other circuitry can be inserted in series with the rotor coils through carbon brushes that slide on the slip-rings allowing an electrical connection with the rotating coils. This basically is the difference in construction between a squirrel cage and slip-ring motors. These are helpful in adding external resistors and contactors. The slip necessary to generate the maximum torque (pull-out torque) is directly proportional to the rotor resistance. In the slip-ring motor, the effective rotor resistance is increased by adding external resistance through the slip rings. Thus, it is possible to get higher slip and hence, the pull-out torque at a lower speed. A particularly high resistance can result in the pull-out torque occurring at almost zero speed, providing a very high pull-out torque at a low starting current. As the motor accelerates, the value of the resistance can be reduced, altering the motor characteristic to suit the load requirement. Once the motor reaches the base speed, external resistors are removed from the rotor. This means that now the motor is working as the standard induction motor.

This motor type is ideal for very high inertia loads, where it is required to generate the pull-out torque at almost zero speed and accelerate to full speed in the minimum time with minimum current draw.

Modifying the speed torque curve by altering the rotor resistors, the speed at which the motor will drive a particular load can be altered. At full load the speed can be reduced effectively to about 50% of the motor synchronous speed, particularly when driving variable torque/variable speed loads, such as printing presses, compressors, conveyer belts, hoists and elevators. Reducing the speed below 50%, results in very low efficiency due to higher power dissipation in the rotor resistances. This type of motor is used in applications for driving variable torque/ variable speed loads.

Direct-Current Motors

Direct-Current motors, as the name implies, use direct-unidirectional, current. Direct current motors are used in special applications- where high torque starting or where smooth acceleration over a broad speed range is required.

Synchronous Motors

AC power is fed to the stator of the synchronous motor. The rotor is fed by DC from a separate source. The rotor magnetic field locks onto the stator rotating magnetic field and rotates at the same speed. The speed of the rotor is a function of the supply frequency and the number of magnetic poles in the stator. While induction motors rotate with a slip, i.e., rpm is less than the synchronous speed, the synchronous motor rotate with no slip, i.e., the RPM is same as the synchronous speed governed by supply frequency and number of poles. The slip energy is provided by the D.C. excitation power.

Permanent Magnet Synchronous Motor (PMSM)

The permanent magnet synchronous motor (PMSM) is an alternative for AC induction motors due to various advantages such as power density, better cooling, smaller size, better efficiency and so on. These PMSM's have rotor structures similar to brushless DC motors which contain permanent magnets. However, their stator structure is similar to Induction Motor wherein the windings are assembled such that they produce a sinusoidal flux density in the air gap of the motor. As a result, these motors perform best when driven by sinusoidal waveforms.

Synchronous Reluctance Motors

A synchronous reluctance motor has the same structure as that of a salient pole synchronous motor except that it does not have a field winding on the rotor. These motors are becoming popular due its superior performance and capable of achieving IE4 efficiency class. Synchronous reluctance Motors Stator is similar to induction motors and permanent magnet synchronous motors (PMSMs) and its rotor is built with simple magnetic materials to take advantage of the reluctance principle. The Synchronous reluctance motor rotor runs at synchronous speed and there are no magnets or current-conducting parts in the rotor. Hence rotor losses are very small compared to those of an induction motor.

2.3 Motor Characteristics

Motor Speed

The speed of a motor is the number of revolutions in a given time frame, typically revolutions per minute (RPM). The speed of an AC motor depends on the frequency of the input power and the number of poles for which the motor is wound. The synchronous speed in RPM is given by the following equation, where the frequency is in hertz or cycles per second:

$$\text{Synchronous Speed (RPM)} = \frac{120 \times \text{Frequency}}{\text{No. of Poles}}$$

Indian motors have synchronous speeds like 3000 / 1500 / 1000 / 750 / 600 / 500 / 375 RPM corresponding to no. of poles being 2, 4, 6, 8, 10, 12, 16 (always even) and given the mains frequency of 50 cycles / sec.

The actual speed, with which the motor operates, will be less than the synchronous speed. The difference between synchronous and full load speed is called slip and is measured in percent. It is calculated using this equation:

$$\text{Slip (\%)} = \frac{\text{Synchronous Speed - Full Load Rated Speed}}{\text{Synchronous Speed}} \times 100$$

As per relation stated above, the speed of an AC motor is determined by the number of motor poles and by the input frequency. It can also be seen that theoretically speed of an AC motor can be varied infinitely by changing the frequency. Manufacturer's guidelines should be referred for practical limits

to speed variation. With the addition of a Variable Frequency Drive (VFD), the speed of the motor can be decreased as well as increased.

Power Factor

The power factor of the motor is given as: Power Factor = Cos ϕ = $\frac{kW}{kVA}$

As the load on the motor comes down, the magnitude of the **active current** reduces. However, there is no corresponding reduction in the **magnetizing current,** which is proportional to supply voltage with the result that the motor power factor reduces, with a reduction in applied load. Induction motors, especially those operating below their rated capacity, are the main reason for low power factor in electric systems.

2.4 Motor Efficiency

Two important attributes relating to efficiency of electricity use by A.C. Induction motors are efficiency (η), defined as the ratio of the mechanical energy delivered at the rotating shaft to the electrical energy input at its terminals, and power factor (PF). Motors, like other inductive loads, are characterized by power factors less than one. As a result, the total current draw needed to deliver the same real power is higher than for a load characterized by a higher PF. An important effect of operating with a PF less than one is that resistance losses in wiring upstream of the motor will be higher, since these are proportional to the square of the current. Thus, both a high value for η and a PF close to unity are desired for efficient overall operation in a plant.

Squirrel cage motors are normally more efficient than slip-ring motors, and higher-speed motors are normally more efficient than lower-speed motors. Efficiency is also a function of motor temperature. Totally-enclosed, fan-cooled (TEFC) motors are more efficient than screen-protected, drip-proof (SPDP) motors. Also, as with most equipment, motor efficiency increases with the rated capacity.

The efficiency of a motor is determined by intrinsic losses that can be reduced only by changes in motor design. Intrinsic losses are of two types: **fixed losses-** independent of motor load, and **variable losses -** dependent on load.

Fixed losses consist of magnetic core losses and friction and windage losses. Magnetic core losses (sometimes called iron losses) consist of eddy current and hysteresis losses in the stator. They vary with the core material and geometry and with input voltage.

Friction and windage losses are caused by friction in the bearings of the motor and aerodynamic losses associated with the ventilation fan and other rotating parts.

Variable losses consist of resistance losses in the stator and in the rotor and miscellaneous stray losses. Resistance to current flow in the stator and rotor result in heat generation, that is proportional to the resistance of the material and the square of the current (I^2R). Stray losses arise from a variety of sources and are difficult to either measure directly or to calculate, but are generally proportional to the square of the rotor current.

Part-load performance characteristics of a motor also depend on its design. Both the η and PF fall to very low levels at low loads. The Figures 2.2 shows the effect of load on power factor and efficiency. It can be seen that power factor drops sharply at part loads. The Figure 2.3 shows the effect of speed on power factor.

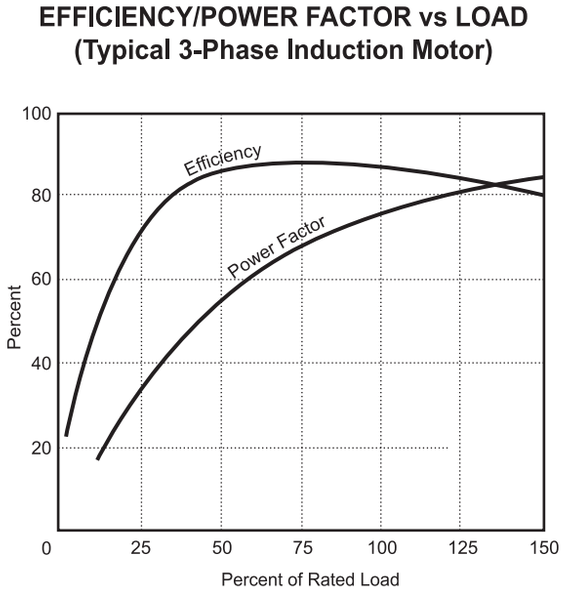

Figure 2.2 % Load vs. Power factor, Efficiency

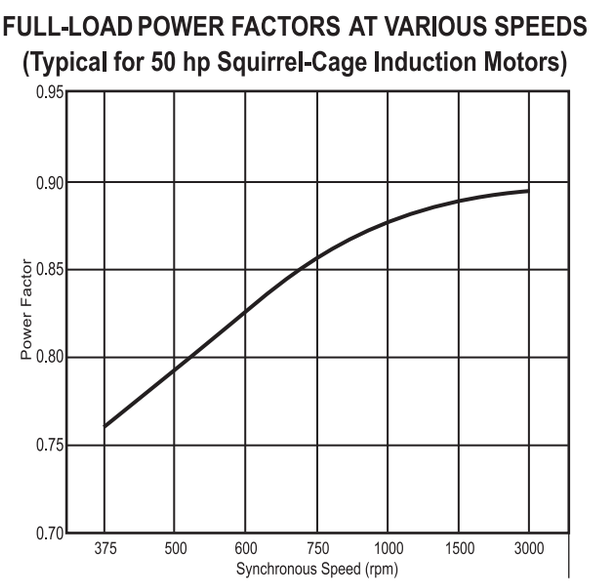

Figure 2.3 Speed vs. Power factor

Field Tests for Determining Efficiency

The efficiency of the motor is given by

$$\eta = \frac{P_{out}}{P_{in}} = 1 - \frac{P_{loss}}{P_{in}}$$

Where P_{out} = Output power of the motor

P_{in} = Input power of the motor

P_{loss} = Losses occurring in motor

The various losses in the motor are determined as follows:

No Load Test: The motor is run at rated voltage and frequency without any shaft load. Input power, current, frequency and voltage are noted. The no load P.F. is quite low and hence low PF watt meters are required. From the input power, stator I^2R losses under no load are subtracted to give the sum of Friction and Windage (F&W) and core losses. To separate core and F & W losses, test is repeated at variable voltages. It is useful to plot no-load input kW versus Voltage; the intercept is Friction & Windage kW loss component.

F&W and core losses = No load power (Watts) – (No load current)^2 × Stator resistance

Stator and Rotor I^2R Losses:

The stator winding resistance is directly measured by a bridge or volt amp method. The resistance must be corrected to the operating temperature. For modern motors, the operating temperature is likely to be in the range of 100^0C to 120^0C and necessary correction should be made. Correction to 75^0C may be inaccurate. The correction factor is given as follows:

$$\frac{R_2}{R_1} = \frac{235 + t_2}{235 + t_1}$$, where, t_1 = ambient temperature, ^{o}C & t_2 = operating temperature, ^{o}C.

The rotor resistance can be determined from locked rotor test, but rotor I^2R losses are measured from measurement of rotor slip.

Rotor I^2R losses = Slip × (Stator Input – Stator I^2R Losses – Core Loss)

Accurate measurement of slip is possible by stroboscope or non-contact type tachometer. Slip also must be corrected to operating temperature.

Stray Load Losses:

These losses are difficult to measure with any accuracy. IEEE Standard 112 gives a complicated method, which is rarely used on shop floor. IS and IEC standards take a fixed value as 0.5 % of input. The actual value of stray losses is likely to be more. IEEE – 112 specifies values from 0.9 % to 1.8 % (see Table 2.1.)

Table 2.1 Motor Rating Vs. Stray Losses - IEEE

Motor Rating	Stray Losses
1 – 125 HP	1.8 %
125 – 500 HP	1.5 %
501 – 2499 HP	1.2 %
2500 and above	0.9 %

Pointers for Users:

It must be clear that accurate determination of efficiency is very difficult. The same motor tested by different methods and by same methods by different manufacturers can give a difference of 2 %. In view of this, for selecting high efficiency motors, the following can be done:

a. When purchasing large number of small motors or a large motor, ask for a detailed test certificate. If possible, try to remain present during the tests; this will add cost.

b. See that efficiency values are specified without any tolerance

c. Check the actual input current and kW, if replacement is done

d. For new motors, keep a record of no load input power and current

e. Use values of efficiency for comparison and for confirming; rely on measured inputs for all calculations.

Estimation of efficiency in the field can be done as follows:

a. Measure stator resistance and correct to operating temperature. From rated current value, I^2R losses are calculated.

b. From rated speed and output, rotor I^2R losses are calculated

c. From no load test, core and F & W losses are determined for stray loss

The method is illustrated by the following example:

Motor Specifications

Rated power	=	34 kW/45 HP
Voltage	=	415 Volt
Current	=	57 Amps
Speed	=	1475 rpm
Insulation class	=	F
Frame	=	LD 200 L
Connection	=	Delta

No load test Data

Voltage, V	=	415 Volts
Current, I	=	16.1 Amps
Frequency, F	=	50 Hz
Stator phase resistance at 30^0C	=	0.264 Ohms
No load power, P_{nl}	=	1063.74 Watts

a. Calculate iron plus friction and windage losses

b. Calculate stator resistance at 120^0C

$$R_2 = R_1 \times \frac{235 + t_2}{235 + t_1}$$

c. Calculate stator copper losses at operating temperature of resistance at 120^0C

d. Calculate full load slip(s) and rotor input assuming rotor losses are slip times rotor input.

e. Determine the motor input assuming that stray losses are 0.5 % of the motor rated power

f. Calculate motor full load efficiency and full load power factor

Solution

a) Let Iron plus friction and windage loss, P_i + fw
No load power, P_{nl} = 1063.74 Watts
Stator Copper loss, P st-30^0C (Pst.cu)
$= 3 \times (16.1 / \sqrt{3})^2 \times 0.264$
= 68.43 Watts

$Pi + fw = P_{nl} - Pst.cu$
$= 1063.74 - 68.43$
$= 995.3\ W$

b) Stator Resistance at 120^0C,

$$R_{120\ ^0C} = 0.264 \times \frac{120 + 235}{30 + 235}$$

$= 0.354$ ohms per phase

c) Stator copper losses at full load, Pst.cu 120^0C

$= 3 \times (57 / \sqrt{3})^2 \times 0.354$
$= 1150.1$ Watts

d) Full load slip

$S = (1500 - 1475) / 1500$
$= 0.0167$

Rotor input, Pr $= P_{output} / (1\text{-}S)$
$= 34000 / (1\text{-}0.0167)$
$= 34577.4$ Watts

e) Motor full load input power, P input

$= P_r + Pst.cu\ 120^0C + (P_i + fw) + P_{stray}$
$= 34577.4 + 1150.1 + 995.3 + (0.005^* \times 34000)$
$= 36892.8$ Watts

*where, stray losses = 0.5% of rated output (assumed)

f) Motor efficiency at full load

$$\text{Efficiency} = \frac{P_{output}}{P_{input}} \times 100$$

$$= \frac{34000}{36892.8} \times 100 = 92.2\ \%$$

$$\text{Full Load PF} = \frac{P_{input}}{\sqrt{3} \times V \times I_{fl}}$$

$$= \frac{36892.8}{\sqrt{3} \times 415 \times 57} = 0.90$$

Comments:

a. The measurement of stray load losses is very difficult and not practical even on test beds.

b. The actual value of stray loss of motors up to 200 HP is likely to be 1 % to 3 % compared to 0.5 % assumed by standards.

c. The value of full load slip taken from the nameplate data is not accurate. Actual measurement under full load conditions will give better results.

d. The friction and windage losses really are part of the shaft output; however, in the above calculation, it is not added to the rated shaft output, before calculating the rotor input power. The error however is minor.

e. When a motor is rewound, there is a fair chance that the resistance per phase would increase due to winding material quality and the losses would be higher. It would be interesting to assess the effect of a nominal 10 % increase in resistance per phase.

2.5 Motor Selection

The primary technical consideration defining the motor choice for any particular application is the torque required by the load, especially the relationship between the maximum torque generated by the motor (break-down torque) and the torque requirements for start-up (locked rotor torque) and during acceleration periods.

The duty / load cycle determines the thermal loading on the motor. One consideration with totally enclosed fan cooled (TEFC) motors is that the cooling may be insufficient when the motor is operated at speeds below its rated value.

Ambient operating conditions affect motor choice; special motor designs are available for corrosive or dusty atmospheres, high temperatures, restricted physical space, etc.

An estimate of the switching frequency (usually dictated by the process), whether automatic or manually controlled, can help in selecting the appropriate motor for the duty cycle.

The demand a motor will place on the balance of the plant electrical system is another consideration - if the load variations are large, for example as a result of frequent starts and stops of large components like compressors, the resulting large voltage drops could be detrimental to other equipment.

Reliability is of prime importance - in many cases, however, designers and process engineers seeking reliability will grossly oversize equipment, leading to sub-optimal energy performance. Good knowledge of process parameters and a better understanding of the plant power system can aid in reducing over sizing with no loss of reliability.

Inventory is another consideration - Many large industries use standard equipment, which can be easily serviced or replaced, thereby reducing the stock of spare parts that must be maintained and minimizing shut-down time. This practice affects the choice of motors that might provide better energy performance in specific applications. Shorter lead times for securing individual motors from suppliers would help reduce the need for this practice.

Price is another issue - Many users are first-cost sensitive, leading to the purchase of less expensive motors that may be more costly on a lifecycle basis because of lower efficiency. For example, energy efficient motors or other specially designed motors typically save within a few years an amount of money equal to several times the incremental cost for an energy efficient motor, over a standard-efficiency motor. Few of salient selection issues are given below:

- In the selection process, the power drawn at 75 % of loading can be a meaningful indicator of energy efficiency.
- Reactive power drawn (kVAr) by the motor.
- Indian Standard 325 for standard motors allows 15 % tolerance on efficiency for motors upto 50 kW rating and 10 % for motors over 50 kW rating.
- The Indian Standard IS 8789 addresses technical performance of Standard Motors while IS 12615 addresses the efficiency criteria of High Efficiency Motors. Both follow IEC 34-2 test methodology wherein, stray losses are assumed as 0.5 % of input power. By the IEC test method, the losses are understated and if one goes by IEEE test methodology, the motor efficiency values would be further lowered.
- It would be prudent for buyers to procure motors based on test certificates rather than labeled values.
- The energy savings by motor replacement can be worked out by the simple relation : kW savings = kW output × [$1/\eta_{old}$ - $1/\eta_{new}$] where η_{old} and η_{new} are the existing and proposed motor efficiency values.
- The cost benefits can be worked out on the basis of premium required for high efficiency vs. worth of annual savings.

2.6 Energy Efficient Motors

Energy-efficient motors (EEM), are the ones in which, design improvements are incorporated specifically to increase operating efficiency over motors of standard design (see figure 2.4). Design improvements focus on reducing intrinsic motor losses. Improvements include the use of lower-loss silicon steel, a longer core (to increase active material), thicker wires (to reduce resistance), thinner laminations, smaller air gap between stator and rotor, copper instead of aluminum bars in the rotor, superior bearings and a smaller fan, etc.

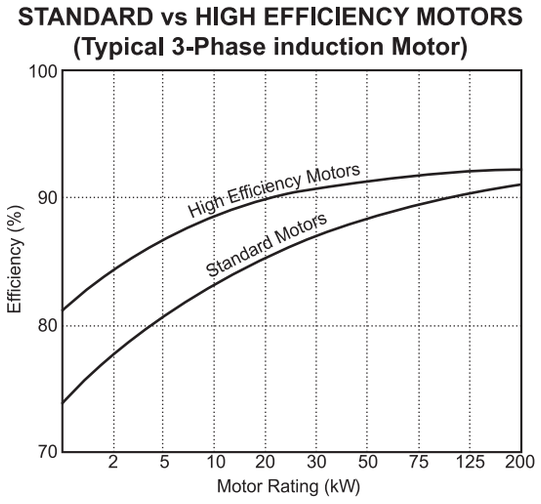

Figure 2.4 Standard vs High Efficiency Motors

Energy-efficient motors now available in India operate with efficiencies that are typically 3 to 4 percentage points higher than standard motors. In keeping with the stipulations of the BIS, energy-efficient motors are designed to operate without loss in efficiency at loads between 75 % and 100 % of rated capacity. This may result in major benefits in varying load applications. The power factor is about the same or may be higher than for standard motors. Furthermore, energy-efficient motors have lower operating temperatures and noise levels, greater ability to accelerate higher-inertia loads, and are less affected by supply voltage fluctuations.

Minimising Watts Loss in Motors

Improvements in motor efficiency can be achieved without compromising motor performance - at higher cost - within the limits of existing design and manufacturing technology.

From the Table 2.2, it can be seen that any improvement in motor efficiency must result from reducing the Watts losses. In terms of the existing state of electric motor technology, a reduction in watts losses can be achieved in various ways.

Table 2.2 Energy Efficient Motors

Power Loss Area	Efficiency Improvement
1. Iron	Use of thinner gauge, lower loss core steel reduces eddy current losses. Longer core adds more steel to the design, which reduces losses due to lower operating flux densities.
2. Stator I^2R	Use of more copper and larger conductors increases cross sectional area of stator windings. This lowers resistance (R) of the windings and reduces losses due to current flow (I).
3. Rotor I^2R	Use of larger rotor conductor bars increases size of cross section, lowering conductor resistance (R) and losses due to current flow (I).
4. Friction & Windage	Use of low loss fan design reduces losses due to air movement.
5. Stray Load Loss	Use of optimized design and strict quality control procedures minimizes stray load losses.

Stator and Rotor I^2R Losses

These losses are major losses and typically account for 55% to 60% of the total losses. I^2R losses are heating losses resulting from current passing through stator and rotor conductors. I^2R losses are the function of a conductor resistance, the square of current. Resistance of conductor is a function of conductor material, length and cross sectional area. The suitable selection of copper conductor size will reduce the resistance. Reducing the motor current is most readily accomplished by decreasing the magnetizing component of current. This involves lowering the operating flux density and possible shortening of air gap. Rotor I^2R losses are a function of the rotor conductors (usually aluminium) and the rotor slip. Utilisation of copper conductors will reduce the winding resistance. Motor operation closer to synchronous speed will also reduce rotor I^2R losses.

Core Losses

Core losses are those found in the stator-rotor magnetic steel and are due to hysterisis effect and eddy current effect during 50 Hz magnetization of the core material. These losses are independent of load and account for 20 – 25 % of the total losses.

The hysteresis losses which are a function of flux density, are be reduced by utilizing low-loss grade of silicon steel laminations. The reduction of flux density is achieved by suitable increase in the core

length of stator and rotor. Eddy current losses are generated by circulating current within the core steel laminations. These are reduced by using thinner laminations.

Friction and Windage Losses

Friction and windage losses results from bearing friction, windage and circulating air through the motor and account for 8 – 12 % of total losses. These losses are independent of load. The reduction in heat generated by stator and rotor losses permits the use of smaller fan. The windage losses also reduce with the diameter of fan leading to reduction in windage losses.

Stray Load-Losses

These losses vary according to square of the load current and are caused by leakage flux induced by load currents in the laminations and account for 4 to 5 % of total losses. These losses are reduced by careful selection of slot numbers, tooth/slot geometry and air gap.

Energy efficient motors cover a wide range of ratings and the full load efficiencies are higher by 3 to 7 %. The mounting dimensions are also maintained as per IS1231 to enable easy replacement.

As a result of the modifications to improve performance, the costs of energy-efficient motors are higher than those of standard motors. The higher cost will often be paid back rapidly in saved operating costs, particularly in new applications or end-of-life motor replacements. In cases where existing motors have not reached the end of their useful life, the economics will be less clearly positive.

Because the favourable economics of energy-efficient motors are based on savings in operating costs, there may be certain cases which are generally economically ill-suited to energy-efficient motors. These include highly intermittent duty or special torque applications such as hoists and cranes, traction drives, punch presses, machine tools, and centrifuges. In addition, energy, efficient designs of multi-speed motors are generally not available. Furthermore, energy-efficient motors are not yet available for many special applications, e.g. for flame-proof operation in oil-field or fire pumps or for very low speed applications (below 750 rpm). Also, most energy-efficient motors produced today are designed only for continuous duty cycle operation.

Given the tendency of over sizing on the one hand and ground realities like; voltage, frequency variations, efficacy of rewinding in case of a burnout, on the other hand, benefits of EEM's can be achieved only by careful selection, implementation, operation and maintenance efforts of energy managers.

Technical aspects of Energy Efficient Motors

Energy-efficient motors last longer, and may require less maintenance. At lower temperatures, bearing grease lasts longer; required time between re-greasing increases. Lower temperatures translate to long lasting insulation. Generally, motor life doubles for each 10°C reduction in operating temperature.

Select energy-efficient motors with a 1.15 service factor, and design for operation at 85% of the rated motor load.

Electrical power problems, especially poor incoming power quality can affect the operation of energy-efficient motors.

Speed control is crucial in some applications. In polyphase induction motors, slip is a measure of motor winding losses. Lower the slip, higher the efficiency. Less slippage in energy efficient motors results in speeds about 1% faster than in standard counterparts.

Starting torque for efficient motors may be lower than for standard motors. Facility managers should be careful when applying efficient motors to high torque applications.

2.7 Factors Affecting Energy Efficiency & Minimising Motor Losses in Operation

Power Supply Quality

Motor performance is affected considerably by the quality of input power that is the actual volts and frequency available at motor terminals vis-à-vis rated values as well as voltage and frequency variations and voltage unbalance across the three phases. Motors in India must comply with standards set by the Bureau of Indian Standards (BIS) for tolerance to variations in input power quality. The BIS standards specify that a motor should be capable of delivering its rated output with a voltage variation of +/- 6 % and frequency variation of +/- 3 %. Fluctuations much larger than these are quite common in utility-supplied electricity in India. Voltage fluctuations can have detrimental impacts on motor performance. The general effects of voltage and frequency variation on motor performance are presented in Table 2.3:

Table 2.3 General Effects of Voltage and Frequency Variation on Induction Motor Characteristics

	Percen-tage	Starting and Max. Running Torque	Syn-chronous Speed %	% Slip	Full-load Speed %	Efficiency			Power Factor			Full-load Current %	Starting Current %	Max. Overload Capacity %
						100 % Load	75 % Load	50 % Load	100 % Load	75 % Load	50 % Load			
Voltage	120	↑ 44	None	↓ 30	↑ 1.5	Small incr.	↓ 5 to 2 pts.	↓ 7 to 20 pts	↓ 5 to 15 pts	↓ 10 to 30 pts	↓ 15 to 40 pts	↓ 11	↑ 25	↑ 44
	110	↑ 21	None	↓ 17	↑ 1	↑ .5 to 1 pt.	Almost none	↓ 1 to 2 pts	↓ 3 pts	↓ 4 pts	↓ 5 to 6 pts	↓ 7	↑ 10 to 12	↑ 21
	90	↓ 19	None	↑ 23	↓ 1.5	↓ 2 pts.	Almost none	↑ 1 to 2 pts	↑ 1 pt	↑ 2 to 3 pts	↑ 4 to 5 pts	↑11	↓ 10 to 12	↓ 19
	Function of	$(volt)^2$	Const.	$(volt)^2$	(Synch. speed slip)								(volt.)	$(volt.)^2$
Frequency	105	↓ 10	↑ 5	Almost none	↑ 5	---------- slight increase each load ----------						Slight	↓ 5 to 6	Slight ↓
	95	↑ 11	↓ 5	Almost non	↓ 5	---------- slight decrease each load ----------						Slight ↑	↑ 5 to 6	Slight ↑
	Function of	$(freq.)^2$	(freq.)		(synch. Speed slip)								$(freq.)^{-1}$	

Note : pts. Indicate the change in value and not as percentage, for example, 2 pts reduction from 75% efficiency means the new efficiency will be 73% .

The options available for an energy manager to ensure near to rated voltage at motor terminals include:

i) Load end power factor improvement by providing matching PF capacitors

ii) Minimizing line / cable voltage drops from sub-station to motor terminals

iii) Transformer tap changing as required in case of consistent and continuous low voltage situations.

Voltage Unbalance

Voltage unbalance, the condition where the voltages in the three phases are not equal, can be still more detrimental to motor performance and motor life. Unbalance typically occurs as a result of supplying single-phase loads disproportionately from one of the phases. It can also result from the use of different sizes of cables in the distribution system. An example of the effect of voltage unbalance on motor performance is shown in Table 2.4.

Table 2.4 Example of the Effect of Voltage Unbalance on Motor Performance

Parameter	Percent unbalance in voltage		
	0.30	2.30	5.40
Unbalance in current (%)	0.4	17.7	40.0
Increased temperature rise (^{0}C)	0.18	10.6	58

The NEMA (National Electrical Manufacturers Association of USA) standard definition of voltage unbalance is given by the following equation:

$$Voltage\ unbalance = \frac{Maximum\ deviation\ from\ mean\ of\ V_{ab}, V_{bc}, V_{ca}}{Mean\ of\ (V_{ab}, V_{bc}, V_{ca})}$$

As an example, consider a three-phase supply system (in volts):

The line-line voltages are:

$V_{ab} = 410$ $V_{bc} = 417$ $V_{ca} = 408$

% Voltage Unbalance = (417 – 411.7/ 411.667) x 100 = 1.29 %

Where

Mean = (410 + 417 + 408) / 3 = 411.7

Hence the voltage unbalance is 1.29%.

Common Causes of Voltage Unbalance

It is recommended that the voltage unbalance at the motor terminals not exceed 1% , anything above this will lead to derating of the motor. The common causes of voltage unbalance are

Some of the more common causes of unbalanced voltages are:

- Unbalanced incoming utility supply
- Unequal transformer tap settings
- Large single phase distribution transformer on the system
- Open phase on the primary of a 3 phase transformer on the distribution system
- Faults or grounds in the power transformer

- Open delta connected transformer banks
- A blown fuse on a 3 phase bank of power factor improvement capacitors
- Unequal impedance in conductors of power supply wiring
- Unbalanced distribution of single phase loads such as lighting
- Heavy reactive single phase loads such as welders

Voltage unbalance is probably the leading power factor problem that results in motor over heating and premature motor failure.

Voltage unbalance causes extremely high current imbalance. The magnitude of current imbalance may be 6 to 10 times as large as the voltage imbalance. A motor will run hotter when operating on a power supply with voltage unbalance. The additional temperature rise is estimated with the following equation

$$\text{Additional temperature rise} = 2 \times (\%\ \text{Voltage unbalance})^2$$

For example, if the voltage unbalance is 2% for a motor operating at 100 ^{o}C, the additional temperature rise will be 8 ^{o}C. The winding insulation life is reduced by one half for each 10 ^{o}C increase in operating temperature.

Motor Loading

Measuring Load

% Loading of the motor can be estimated by the following relation:

$$\%\ Loading = \frac{Input\ power\ drawn\ by\ the\ motor\ (kW)\ at\ existing\ load}{Name\ plate\ full\ load\ kW\ rating\ /\ name\ plate\ full\ load\ motor\ efficiency} x100$$

(or)

$$\%\ Loading = \frac{Input\ power\ drawn\ by\ the\ motor\ (kW)\ at\ existing\ load}{\sqrt{3}\ x\ kV\ x\ I\ x\ Cos\phi}\ x\ 100$$

- Never assume power factor
- Loading should not be estimated as the ratio of currents.

$$Motor\ Loading\ \% = \frac{Actual\ operating\ load\ of\ the\ motor}{Rated\ capacity\ of\ the\ motor}\ x\ 100$$

Motor Load Survey: Methodology

Large industries have a massive population of LT motors. Load survey of LT motors can be taken-up methodically to identify improvement options as illustrated in following case study.

i) Sampling Criteria

Towards the objective of selecting representative LT motor drives among the motor population, for analysis, the criteria considered are:

- Utilization factor i.e., hours of operation with preference given to continuously operated drive motors.
- Sample representative basis, where one drive motor analysis can be reasoned as representative for the population. Ex : Cooling Tower Fans, Air Washer Units, etc.
- Conservation potential basis, where drive motors with inefficient capacity controls on the machine side, fluctuating load drive systems, etc., are looked into.

ii) Measurements

Studies on selected LT motors involve measurement of electrical load parameters namely volts, amperes, power factor, kW drawn.

Observations on machine side parameters such as speed, load, pressure, temperature, etc., (as relevant) are also taken. Availability of online instruments for routine measurements, availability of tail-end capacitors for PF correction, energy meters for monitoring is also looked into for each case.

iii) Analysis

Analysis of observations on representative LT motors and connected drives is carried out towards following outputs:

- Motor load on kW basis and estimated energy consumption.
- Scope for improving monitoring systems to enable sustenance of a regular in-house Energy Audit function.
- Scope areas for energy conservation with related cost benefits and source information.

The observations are to indicate:

% loading on kW, % voltage unbalance if any, voltage, current, frequency, power factor, machine side conditions like load / unload condition, pressure, flow, temperature, damper / throttle operation, whether it is a rewound motor, idle operations, metering provisions, etc.

The findings / recommendations may include:

- Identified motors with less than 50 % loading, 50 – 75 % loading, 75 – 100 % loading, over 100 % loading.
- Identified motors with low voltage / power factor / voltage imbalance for needed improvement measures.
- Identified motors with machine side losses / inefficiencies like idle operations, throttling / damper operations for avenues like automatic controls / interlocks, variable speed drives, etc.

Motor load survey is aimed not only as a measure to identify motor efficiency areas but equally importantly, as a means to check combined efficiency of the motor, driven machine and controller if

any. The margins in motor efficiency may be less than 10 % of consumption often, but the load survey would help to bring out savings in driven machines / systems, which can give 30 – 40 % energy savings.

Reducing Under-loading

Probably the most common practice contributing to sub-optimal motor efficiency is that of under-loading. Under-loading results in lower efficiency and power factor, and higher-than-necessary first cost for the motor and related control equipment. Under-loading is common for several reasons. Original equipment manufacturers tend to use a large safety factor in motors they select. Under-loading of the motor may also occur from under-utilisation of the equipment. For example, machine tool equipment manufacturers provide for a motor rated for the full capacity load of the equipment ex. depth of cut in a lathe machine. The user may need this full capacity rarely, resulting in under-loaded operation most of the time. Another common reason for under-loading is selection of a larger motor to enable the output to be maintained at the desired level even when input voltages are abnormally low. Finally, under-loading also results from selecting a large motor for an application requiring high starting torque where a special motor, designed for high torque, would have been suitable.

A careful evaluation of the load would determine the capacity of the motor that should be selected. Another aspect to consider is the incremental gain in efficiency achievable by changing the motor. Larger motors have inherently higher rated efficiencies than smaller motors. Therefore, the replacement of motors operating at 60 – 70 % of capacity or higher is generally not recommended. However, there are no rigid rules governing motor selection; the savings potential needs to be evaluated on a case-to-case basis. When downsizing, it may be preferable to select an energy-efficient motor, the efficiency of which may be higher than that of a standard motor of higher capacity.

Improving the Motor Loading by Operating in Star Mode

For motors, which consistently operate at loads below 40 % of rated capacity, an inexpensive and effective measure might be to operate in star mode. A change from the standard delta operation to permanent star operation involves re-configuring the wiring at terminal box and resetting of the over current relay.

Operating in the star mode leads to a voltage reduction by a factor of '$\sqrt{3}$'. Motor is electrically downsized by 1/3rd in star mode operation, but performance characteristics as a function of load remain unchanged. For example if a motor is rated for 15 kW in delta mode, its derated capacity is 5kW in star mode. Thus, full-load operation in star mode gives higher efficiency and power factor than partial load operation in the delta mode. However, motor operation in the star mode is possible only for applications where the torque-to-speed requirement is lower at reduced load.

As speed of the motor reduces in star mode this option may be avoided in case the motor is connected to a production facility whose output is related to the motor speed. Further in star mode the motor loading should not be allowed to cross derated capacity. For example in above case of 15 kW delta connected electric motor, should not be loaded above 5 kW when delta to star switchover takes place.

For applications with high initial torque and low running torque needs, automatic Star-Del-Star converters are also available, which help in load following de-rating of electric motors after initial start-up.

Sizing to Variable Load

Industrial motors frequently operate under varying load conditions due to process requirements. A common practice in cases where such variable-loads are found is to select a motor based on the highest anticipated load. In many instances, an alternative approach is typically less costly, more efficient, and provides equally satisfactory operation. With this approach, the optimum rating for the motor is selected on the basis of the load duration curve for the particular application. Thus, rather than selecting a motor of high rating that would operate at full capacity for only a short period, a motor would be selected with a rating slightly lower than the peak anticipated load and would operate at overload for a short period of time. Since operating within the thermal capacity of the motor insulation is of greatest concern in a motor operating at higher than its rated load, the motor rating is selected as that which would result in the same temperature rise under continuous full-load operation as the weighted average temperature rise over the actual operating cycle. Under extreme load changes, e.g. frequent starts / stops, or high inertial loads, this method of calculating the motor rating is unsuitable since it would underestimate the heating that would occur.

Where loads vary substantially with time, in addition to proper motor sizing, the control strategy employed can have a significant impact on motor electricity use. Traditionally, mechanical means (e.g. throttle valves in piping systems) have been used when lower output is required. More efficient speed control mechanisms include multi-speed motors, eddy-current couplings, fluid couplings, and solid-state electronic variable speed drives.

Power Factor Correction

As noted earlier, induction motors are characterized by power factors less than unity, leading to lower overall efficiency (and higher overall operating cost) associated with a plant's electrical system. Capacitors connected in parallel (shunted) with the motor are typically used to improve the power factor. The impacts of PF correction include reduced kVA demand (and hence reduced utility demand charges), reduced I^2R losses in cables upstream of the capacitor (and hence reduced energy charges), reduced voltage drop in the cables (leading to improved voltage regulation), and an increase in the overall efficiency of the plant electrical system.

It should be noted that PF capacitor improves power factor from the point of installation back to the generating side. It means that, if a PF capacitor is installed at the starter terminals of the motor, it won't improve the operating PF of the motor, but the PF from starter terminals to the power generating side will improve, i.e., the benefits of PF would be only on upstream side.

The size of capacitor required for a particular motor depends upon the no-load reactive kVA (kVAR) drawn by the motor, which can be determined only from no-load testing of the motor. In general, the capacitor is then selected to not exceed 90 % of the no-load kVAR of the motor. (Higher capacitors could result in over-voltages and motor burn-outs). Alternatively, typical power factors of standard motors can provide the basis for conservative estimates of capacitor ratings to use for different size motors. The capacitor rating for power connection by direct connection to induction motors is shown in Table 2.5.

Table 2.5 Capacitor Ratings for Power Factor Correction by Direct Connection to Induction Motors						
Motor Rating (HP)	**Capacitor rating (kVAr) for Motor Speed**					
	3000	**1500**	**1000**	**750**	**600**	**500**
5	2	2	2	3	3	3
7.5	2	2	3	3	4	4
10	3	3	4	5	5	6
15	3	4	5	7	7	7
20	5	6	7	8	9	10
25	6	7	8	9	9	12
30	7	8	9	10	10	15
40	9	10	12	15	16	20
50	10	12	15	18	20	22
60	12	14	15	20	22	25
75	15	16	20	22	25	30
100	20	22	25	26	32	35
125	25	26	30	32	35	40
150	30	32	35	40	45	50
200	40	45	45	50	55	60
250	45	50	50	60	65	70

From the above table, it may be noted that required capacitive kVAr increases with decrease in speed of the motor, as the magnetizing current requirement of a low speed motor is more in comparison to the high speed motor for the same HP of the motor. Since a reductions in line current, and associated energy efficiency gains, are reflected backwards from the point of application of the capacitor, the maximum improvement in overall system efficiency is achieved when the capacitor is connected across the motor terminals, as compared to somewhere further upstream in the plant's electrical system. However, economies of scale associated with the cost of capacitors and the labor required to install them will place an economic limit on the lowest desirable capacitor size.

Maintenance

Inadequate maintenance of motors can significantly increase losses and lead to unreliable operation. For example, improper lubrication can cause increased friction in both the motor and associated drive transmission equipment. Resistance losses in the motor, which rise with temperature, would increase. Providing adequate ventilation and keeping motor cooling ducts clean can help dissipate heat to reduce excessive losses. The life of the insulation in the motor would also be longer : for every 10^0C increase in motor operating temperature over the recommended peak, the time before rewinding would be needed is estimated to be halved.

A checklist of good maintenance practices to help insure proper motor operation would include:

- Inspecting motors regularly for wear in bearings and housings (to reduce frictional losses) and for dirt/dust in motor ventilating ducts (to ensure proper heat dissipation).

- Checking load conditions to ensure that the motor is not over or under loaded. A change in motor load from the last test indicates a change in the driven load, the cause of which should be understood.
- Lubricating appropriately. Manufacturers generally give recommendations for how and when to lubricate their motors. Inadequate lubrication can cause problems, as noted above. Over-lubrication can also create problems, e.g. excess oil or grease from the motor bearings can enter the motor and saturate the motor insulation, causing premature failure or creating a fire risk.
- Checking periodically for proper alignment of the motor and the driven equipment. Improper alignment can cause shafts and bearings to wear quickly, resulting in damage to both the motor and the driven equipment.
- Ensuring that supply wiring and terminal box are properly sized and installed. Inspect regularly the connections at the motor and starter to be sure that they are clean and tight.

Age

Most motor cores in India are manufactured from silicon steel or de-carbonized cold-rolled steel, the electrical properties of which do not change measurably with age. However, poor maintenance (inadequate lubrication of bearings, insufficient cleaning of air cooling passages, etc.) can cause a deterioration in motor efficiency over time. Ambient conditions can also have a detrimental effect on motor performance. For example, excessively high temperatures, high dust loading, corrosive atmosphere, and humidity can impair insulation properties; mechanical stresses due to load cycling can lead to misalignment. However, with adequate care, motor performance can be maintained.

2.8 Rewinding Effects on Energy Efficiency

It is common practice in industry to rewind burnt-out motors. The population of rewound motors in some industries exceeds 50 % of the total population. Careful rewinding can sometimes maintain motor efficiency at previous levels, but in most cases, losses in efficiency result. Rewinding can affect a number of factors that contribute to deteriorated motor efficiency: winding and slot design, winding material, insulation performance, and operating temperature. For example, a common problem occurs when heat is applied to strip old windings: the insulation between laminations can be damaged, thereby increasing eddy current losses. A change in the air gap may affect power factor and output torque.

However, if proper measures are taken, motor efficiency can be maintained, and in some cases increased, after rewinding. Efficiency can be improved by changing the winding design, though the power factor could be affected in the process. Using wires of greater cross section, slot size permitting, would reduce stator losses thereby increasing efficiency. However, it is generally recommended that the original design of the motor be preserved during the rewind, unless there are specific, load-related reasons for redesign.

The impact of rewinding on motor efficiency and power factor can be easily assessed if the no-load losses of a motor are known before and after rewinding. Maintaining documentation of no-load losses and no-load speed from the time of purchase of each motor can facilitate assessing this impact. For example, comparison of no load current and stator resistance per phase of a rewound motor with the

original no-load current and stator resistance at the same voltage can be one of the indicators to assess the efficacy of rewinding.

Performance Evaluation of Rewound Motors

Ideally, a comparison should be made of the efficiency before and after a rewinding. A relatively simple procedure for evaluating rewind quality is to keep a log of no-load input current for each motor in the population. This figure increases with poor quality rewinds. A review of the rewind shop's procedure should also provide some indication of the quality of work. When rewinding a motor, if smaller diameter wire is used, the resistance and the I^2R losses will increase.

The monitoring format for rewound motor is given Table 2.6 below:

Table 2.6 Monitoring Format for Rewound Motors

Section	Equipment Code	Motor Code	Motor Type		No Load Current				Starter Resistance/phase		No load loss	
			Sq.Cage	Slip Ring	New Motor		After Rewinding		New	Rewound	New	Rewound
					A	V	A	V			Watts	Watts

2.9 Speed Control of Motors

Traditionally, DC motors have been employed when variable speed capability was desired. By controlling the armature (rotor) voltage and field current of a separately excited DC motor, a wide range of output speeds can be obtained. DC motors are available in a wide range of sizes, but their use is generally restricted to a few low speed, low-to-medium power applications like machine tools and rolling mills because of problems with mechanical commutation at large sizes. Also, they are restricted for use only in clean, non-hazardous areas because of the risk of sparking at the brushes. DC motors are also expensive relative to AC motors.

Because of the limitations of DC systems, AC motors are increasingly the focus for variable speed applications. Both AC synchronous and induction motors are suitable for variable speed control. Induction motors are generally more popular, however, because of their ruggedness and lower maintenance requirements. AC induction motors are inexpensive (half or less of the cost of a DC motor) and also provide a high power to weight ratio (about twice that of a DC motor).

An induction motor is an asynchronous motor, the speed of which can be varied by changing the supply frequency. The control strategy to be adopted in any particular case will depend on a number of factors including investment cost, load reliability and any special control requirements. Thus, for any particular application, a detailed review of the load characteristics, historical data on process flows, the features required of the speed control system, the electricity tariffs and the investment costs would be a prerequisite to the selection of a speed control system.

The characteristics of the load are particularly important. Load refers essentially to the torque output and corresponding speed required. Loads can be broadly classified as either constant power or constant torque. Constant torque loads are those for which the output power requirement may vary with the speed of operation but the torque does not vary. Conveyors, rotary kilns, and constant-displacement pumps are typical examples of constant torque loads. Variable torque loads are those for which the torque required varies with the speed of operation. Centrifugal pumps and fans are typical examples of variable torque loads (torque varies as the square of the speed). Constant power loads are those for which the torque requirements typically change inversely with speed. Machine tools are a typical example of a constant power load.

The largest potential for electricity savings with variable speed drives is generally in variable torque applications, for example centrifugal pumps and fans, where the power requirement changes as the cube of speed. Constant torque loads are also suitable for VSD application.

Motor Speed Control Systems

Multi-speed motors

Motors can be wound such that two speeds, in the ratio of 2:1, can be obtained. Motors can also be wound with two separate windings, each giving 2 operating speeds, for a total of four speeds. Multi-speed motors can be designed for applications involving constant torque, variable torque, or for constant output power. Multi-speed motors are suitable for applications, which require limited speed control (two or four fixed speeds instead of continuously variable speed), in which cases they tend to be very economical. They have lower efficiency than single-speed motors

Direct Current Drives (DC)

The DC drive technology is the oldest form of electrical speed control. The drive system consists of a DC motor and a controller. The motor is constructed with armature and field windings. Both of these windings require a DC excitation for motor operation. Usually the field winding is excited with a constant level voltage from the controller.

Then, applying a DC voltage from the controller to the armature of the motor will operate the motor. The armature connections are made through a brush and commutator assembly. The speed of the motor is directly proportional to the applied voltage.

The controller is a phase controlled bridge rectifier with logic circuits to control the DC voltage delivered to the motor armature. Speed control is achieved by regulating the armature voltage to the motor. Often a tachogenerator is included to achieve good speed regulation. The tachogenerator would be mounted on the motor and produces a speed feedback signal that is used within the controller.

Wound Rotor AC Motor Drives (Slip Ring Induction Motors)

Wound rotor motor drives use a specially constructed motor to accomplish speed control. The motor rotor is constructed with windings which are brought out of the motor through slip rings on the motor shaft. These windings are connected to a controller which places variable resistors in series with the windings. The torque performance of the motor can be controlled using these variable resistors. Wound rotor motors are most common in the range of 300 HP and above.

Slip Power Recovery Systems

Slip power recovery is a more efficient alternative speed control mechanism for use with slip-ring motors. In essence, a slip power recovery system varies the rotor voltage to control speed, but instead of dissipating power through resistors, the excess power is collected from the slip rings and returned as mechanical power to the shaft or as electrical power back to the supply line. Because of the relatively sophisticated equipment needed, slip power recovery tends to be economical only in relatively high power applications and where the motor speed range is 1:5 or less.

Application of Variable Speed Drives (VSD)

Although there are many methods of varying the speeds of the driven equipment such as hydraulic coupling, gear box, variable pulley etc., the most possible method is one of varying the motor speed itself by varying the frequency and voltage by a variable frequency drive.

Concept of Variable Frequency Drive

The speed of an induction motor is proportional to the frequency of the AC voltage applied to it, as well as the number of poles in the motor stator. This is expressed by the equation:

$$RPM = (f \times 120) / p$$

Where f is the frequency in Hz, and p is the number of poles in any multiple of 2.

Therefore, if the frequency applied to the motor is changed, the motor speed changes in direct proportion to the frequency change. The control of frequency applied to the motor is the job given to the VSD.

The VSD's basic principle of operation is to convert the electrical system frequency and voltage to the frequency and voltage required to drive a motor at a speed other than its rated speed. The two most basic functions of a VSD are to provide power conversion from one frequency to another, and to enable control of the output frequency.

Need for VFD

Earlier motors tended to be over designed to drive a specific load over its entire range. This resulted in a highly inefficient driving system, as a significant part of the input power was not doing any useful work. Most of the time, the generated motor torque was more than the required load torque.

In many applications, the input power is a function of the speed like fan, blower, pump and so on. In these types of loads, the torque is proportional to the square of the speed and the power is proportional to the cube of speed. Variable speed, depending upon the load requirement, provides significant energy saving. A reduction of 20% in the operating speed of the motor from its rated speed will result in an almost 50% reduction in the input power to the motor. This is not possible in a system where the motor is directly connected to the supply line. In many flow control applications, a mechanical throttling device is used to limit the flow. Although this is an effective means of control, it wastes energy because of the high losses and reduces the life of the motor valve due to generated heat.

Principles of VFD's

The VFD is a system made up of active/passive power electronics devices (IGBT, MOSFET, etc.), a high speed central controlling unit and optional sensing devices, depending upon the application requirement. A typical modern-age intelligent VFD for the three phase induction motor is shown in Figure 2.5.

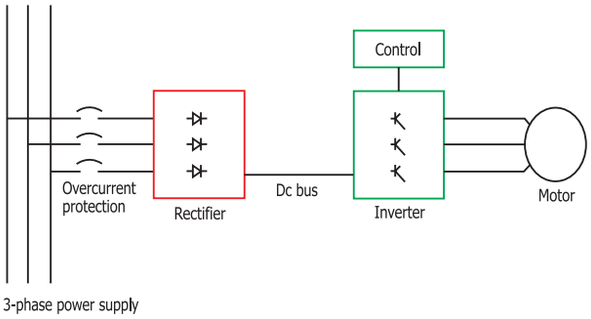

Figure 2.5 Components of a Variable Speed Drive

The basic function of the VFD is to act as a variable frequency generator in order to vary speed of the motor as per the user setting. The rectifier and the filter convert the AC input to DC with negligible ripple. The inverter, under the control of the microcontroller, synthesizes the DC into three-phase variable voltage, variable frequency AC.

The base speed of the motor is proportional to supply frequency and is inversely proportional to the number of stator poles. The number of poles cannot be changed once the motor is constructed. So, by changing the supply frequency, the motor speed can be changed. But when the supply frequency is reduced, the equivalent impedance of electric circuit reduces. This results in higher current drawn by the motor and a higher flux. If the supply voltage is not reduced, the magnetic field may reach the saturation level. Therefore, in order to keep the magnetic flux within working range, both the supply voltage and the frequency are changed in a constant ratio. Since the torque produced by the motor is proportional to the magnetic field in the air gap, the torque remains more or less constant throughout the operating range.

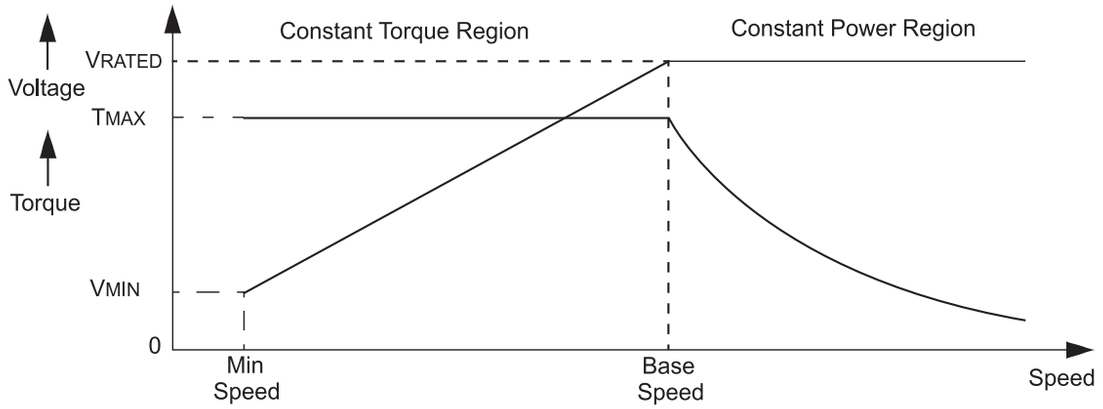

Figure 2. 6 V/f Control

As seen in Figure 2.6, the voltage and the frequency are varied at a constant ratio up to the base speed. The flux and the torque remain almost constant up to the base speed. Beyond the base speed, the supply voltage cannot be increased. Increasing the frequency beyond the base speed results in the field weakening and the torque reduces. Above the base speed, the torque governing factors become more nonlinear as the friction and windage losses increase significantly. Due to this, the torque curve becomes nonlinear. Based on the motor type, the field weakening can go up to twice the base speed. This control is the most popular in industries and is popularly known as the constant V/f control.

By selecting the proper V/f ratio for a motor, the starting current can be kept well under control. This avoids any sag in the supply line, as well as heating of the motor. The VFD also provides overcurrent protection. This feature is very useful while controlling the motor with higher inertia. Since almost constant rated torque is available over the entire operating range, the speed range of the motor becomes wider. User can set the speed as per the load requirement, thereby achieving higher energy efficiency (especially with the load where power is proportional to the cube speed). Continuous operation over almost the entire range is smooth, except at very low speed. This restriction comes mainly due to the inherent losses in the motor, like frictional, windage, iron, etc. These losses are almost constant over the entire speed. Therefore, to start the motor, sufficient power must be supplied to overcome these losses and the minimum torque has to be developed to overcome the load inertia.

A single VFD has the capability to control multiple motors. The VFD is adaptable to almost any operating condition.

VFD Selection

The size of the VFD depends mainly on driven load type and characteristics. This will determine the drive capacity in terms of full load current (FLC) and power delivered (kW).

Driven Load Types and Characteristics

Mechanical load, which is the load on the motor shaft, can be of two types- Constant Torque (CT) or Variable Torque (VT). There is a basic difference between the two loads with respect to load torque variation at different speeds.

A CT load implies that the load torque seen at motor shaft is independent of motor speed.

This means that the load torque remains approximately the same at all speeds. Examples of CT loads include material handling conveyers, reciprocating & screw compressors and certain types of blowers such as roots blower.

A VT load implies that the load torque seen at the motor shaft is dependent upon the motor speed.

Examples of VT loads include centrifugal fans & pumps and centrifugal compressors.

The graphs (Figures 2.7 & 2.8) below describe the torque requirements at various speeds.

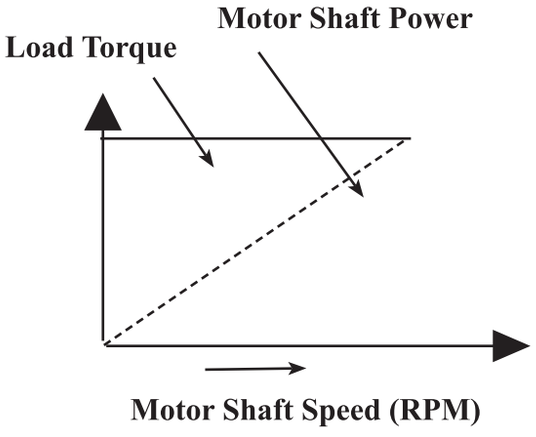

Figure 2.7 A CT load characteristic

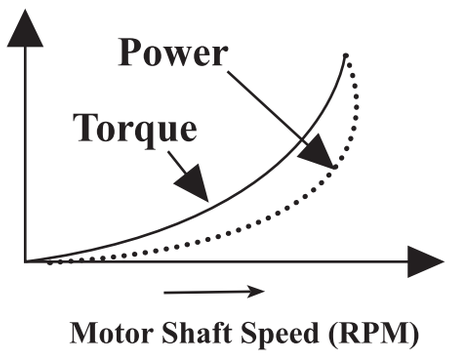

Figure 2.8 A VT load characteristic

Other methods of speed control of motors

In addition to DC drives, VFD and slip ring motors there are other methods used in industries to control the speed of the motors. Some of the common methods used in industries are discussed below:

Eddy Current Drives

This method employs an eddy-current clutch to vary the output speed. The clutch consists of a primary member coupled to the shaft of the motor and a freely revolving secondary member coupled to the load shaft. The secondary member is separately excited using a DC field winding. The motor starts with the load at rest and a DC excitation is provided to the secondary member, which induces eddy-currents in the primary member. The interaction of the fluxes produced by the two currents gives rise to a torque at the load shaft. By varying the DC excitation the output speed can be varied to match the load requirements. The major disadvantage of this system is relatively poor efficiency particularly at low speeds. (See Figure 2.9)

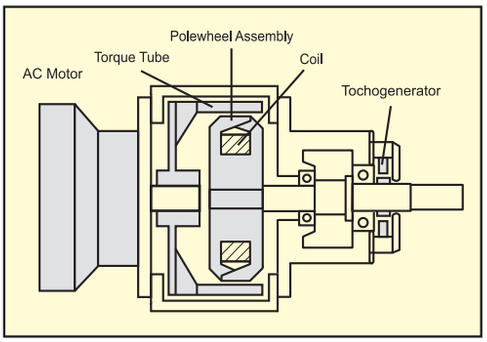

Figure 2.9 Eddy Current Drive

Fluid Coupling

Fluid coupling is one way of applying varying speeds to the driven equipment, without changing the speed of the motor.

Construction

Fluid couplings (see Figure 2.10) work on the hydrodynamic principle. Inside every fluid coupling are two basic elements – the impeller and the runner and together they constitute the working circuit. One can imagine the impeller as a centrifugal pump and the runner as a turbine. The impeller and the rotor are bowl shaped and have large number of radial vanes. They are suitably enclosed in a casing, facing each other with an air gap. The impeller is connected to the prime mover while the rotor has a shaft bolted to it. This shaft is further connected to the driven equipment through a suitable arrangement.

Thin mineral oil of low viscosity and good-lubricating qualities is filled in the fluid coupling from the filling plug provided on its body. A fusible plug is provided on the fluid coupling which blows off and drains out oil from the coupling in case of sustained overloading.

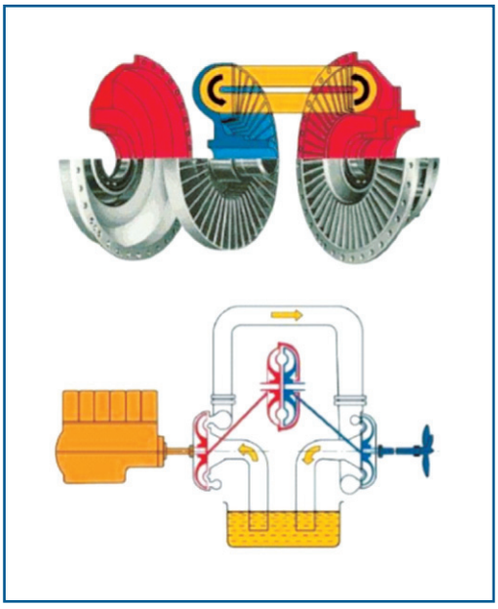

Figure 2.10 Fluid Coupling

Operating Principle

There is no mechanical inter-connection between the impeller and the rotor and the power is transmitted by virtue of the fluid filled in the coupling. When the impeller is rotated by the prime mover, the fluid flows out radially and then axially under the action of centrifugal force. It then crosses the air gap to the runner and is directed towards the bowl axis and back to the impeller. To enable the fluid to flow from impeller to rotor it is essential that there is difference in head between the two and thus it is essential that there is difference in RPM known as slip between the two. Slip is an important and inherent characteristic of a fluid coupling resulting in several desired advantages. As the slip increases, more and more fluid can be transferred. However when the rotor is at a stand still, maximum fluid is transmitted from impeller to rotor and maximum torque is transmitted from the coupling. This maximum torque is the limiting torque. The fluid coupling also acts as a torque limiter.

Characteristics

Fluid coupling has a centrifugal characteristic during starting thus enabling no-load start up of prime mover, which is of great importance. The slipping characteristic of fluid coupling provides a wide range of choice of power transmission characteristics. By varying the quantity of oil filled in the fluid coupling, the normal torque transmitting capacity can be varied. The maximum torque or limiting torque of the fluid coupling can also be set to a pre-determined safe value by adjusting the oil filling. The fluid coupling has the same characteristics in both directions of rotation.

Soft Starter

When starting, AC Induction motor develops more torque than is required at full speed. This stress is transferred to the mechanical transmission system resulting in excessive wear and premature failure of chains, belts, gears, mechanical seals, etc. Additionally, rapid acceleration also has a massive impact on electricity supply charges with high inrush currents drawing +600% of the normal run current.

Figure 2.11 Soft Starter

The use of Star Delta only provides a partial solution to the problem. Should the motor slow down during the transition period, the high peaks can be repeated and can even exceed direct on line current.

Soft starter (see Figure 2.11) provides a reliable and economical solution to these problems by delivering a controlled release of power to the motor, thereby providing smooth, stepless acceleration and deceleration. Motor life will be extended as damage to windings and bearings is reduced.

Soft Start & Soft Stop is built into 3 phase units, providing controlled starting and stopping with a selection of ramp times and current limit settings to suit all applications (see Figure 2.12).

Advantages of Soft Start

- Less mechanical stress
- Improved power factor.
- Lower maximum demand.
- Less mechanical maintenance

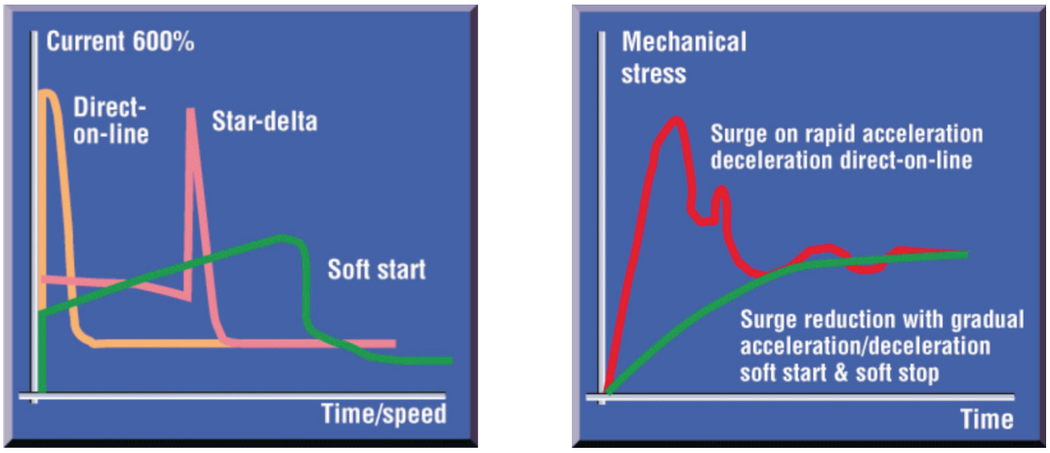

Figure 2.12 Soft Starter: Starting current, Stress profile during starting

2.10 Star Labeling of Energy Efficient Induction Motors

The schedule specifies the requirements for participating in the energy labeling scheme for 3 phase squirrel cage induction motor in 2 Pole, 4 Pole and 6 Pole for continuous duty (S1) operation, suitable for voltage and frequency variation as per IS 12615:2011 having rated output from 0.37 to 375 kW.

In particular, this scheme specifies the following:

1. Rated output (rating)
2. Efficiency Class based on IS 12615:2011 i.e. (IE2, IE2(+), IE3, IE3(+) and IE3 (++))
3. Some of the requirements for energy label validity.
4. The performance criteria for energy labeling validity.
5. Test report format.
6. Label design and details to be incorporated on the label.

Solved Example:

During an energy audit following data were obtained on a 3 phase induction motor:

Rated values: 37 kW,415V, 66 A,0.88 pf

Operating values: 410 V, 49A, 0.76 pf

Note: Motor efficiency in this particular case does not change between 50 –100 % loading.

The plant operates for 7000 hours per year with the electricity cost of Rs. 6.00 per unit.

It is proposed to replace the existing motor by a 30 kW energy efficient motor with 92% efficiency.

a) Determine the rated efficiency and the loading of the existing motor.

b) Calculate the loading with energy efficient motor.

c) If replacing the existing motor with energy efficient motor which costs Rs.75,000, determine the payback period for the investment required for the energy efficient motor over the existing motor. Consider the salvage value of the existing motor as Rs.10,000/.

Ans:

Rated input power	1.732 × 0.415 × 66× 0.88
	41.746 kW
Rated efficiency of the motor	37/ 41.746
	88.63%
Actual input power drawn	1.732 × 0.410 × 49 × 0.76
	26.44 kW
Loading of the motor	26.44/41.746 = 0.633 or 63.3%
Shaft power or motor output	37x0.633= 23.44 kW
Energy efficient motor rating	30 kW
Actual output required	23.44 kW
% loading of the motor	23.44/30
	78 %
Annual energy savings	23.44(1/0.8863 – 1/0.92) x 7000 x Rs.6
	Rs.40,740/-
Payback period	(75,000-10,000)/40740
	1.59 years

QUESTIONS	
Objective Type Questions	
1.	With decrease in speed of the motor, the required capacitive kVAr: a) increases b) decreases c) does not change d) none of the above
2.	Reduction in supply voltage by 10% will change the torque of the motor by a) 38% b) 19% c) 9.5% d) no change
3.	One low investment measure to improve efficiency of a squirrel cage induction motor, which operates consistently below 40% of its rated capacity, is by a) operating it in star mode b) replacing it with a correctly sized motor c) operating in delta mode d) none of the above
4.	In an induction motor, Magnetic field is established in a) stator winding only b) rotor winding only c) stator and Rotor Windings d) none of the above
5.	A 7.5 kW, 415 V, 14.5 A, 1460 RPM, 3 phase rated induction motor with full load efficiency of 88%, draws 10.1 A and 5.1 kW of input power. The percentage loading of the motor is about a) 60 % b) 70 % c) 50% d) none of the above
6.	An induction motor rated for 75 kW and 90 % efficiency, operating at full load, will a) deliver 83.3 kW b) deliver 75 kW c) draw 75 kW d) draw 67.5 kW
7.	Stator phase resistance at 30°C is 0.264 ohms. At 120°C, its value will be a) 0.264 ohms b) 0.354 ohms c) 0.237 ohms d) none of the above
8.	Hydrodynamic principle for speed control is used in a) DC drives b) fluid coupling c) pulse width modulation d) eddy current drive
9.	A four pole induction motor operating at 50 Hz, with 1% slip will run at an actual speed of a) 1500 RPM b) 1515 RPM c) 1485 RPM d) none of the above
10.	kW rating indicated on the name plate of an induction motor indicates a) rated input of the motor b) rated output of the motor c) maximum input power which the motor can draw d) maximum instantaneous input power of the motor
Short Type Questions	
S-1	Calculate the % voltage unbalance if the measured voltages are as follows : $V_{RY} = 425$ $V_{YB} = 418$ $V_{BR} = 423$
S-2	Why is it beneficial to operate motors in star mode for under loaded motors?
S-3	Explain the working of a soft starter and its advantage over other conventional starters.
S-4	How does efficiency loss occur in a rewound motor? How do you check the efficacy of rewound motor?
S-5	List the typical losses in induction motors

Long Type Questions	
L-1	A 7.5 kW, 415 V, 14.5 A, 4 pole, 50 Hz, 3 phase rated squirrel cage induction motor has a full load efficiency and power factor of 89% and 0.88 respectively. An energy auditor measures the following operating data of the motor (a) Supply voltage = 410 V (b) Current drawn = 9.5 A (c) PF = 0.8 (d) Supply frequency = 49.8 Hz (e) RPM = 1480 Find out the following at the motor operating conditions: 1. Power input in kW 2. % motor loading 3. % slip
L-2	An efficiency assessment test was carried out for a standard squirrel cage induction motor in a process plant. The motor specifications are as under. Motor rated specification: 50 HP/ 415 Volt, 60 Amps, 1475 rpm, 3 phase, delta connected The following data was collected during the no load test on the motor. Voltage = 415 Volts Current = 18 Amps Frequency = 50 Hz Stator resistance per phase = 0.27 Ohms No load power = 1080 Watts Ambient temperature = 35 ^{o}C Calculate the following: (i) Iron plus friction and windage losses. (ii) Stator resistance at 120^{o}C. (iii) Stator copper loss at operating temperature at 120^{o}C. (iv) Full load slip and rotor input assuming rotor losses are slip times rotor input. (v) Motor input assuming that stray losses are 0.5% of the motor rated power. (vi) Motor full load efficiency and full load power factor.

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Chapter 3

Compressed Air System

Learning Objectives

In this chapter you will learn about

- Types of compressors
- Comparison of various air compressors
- Compressed air system components
- Performance parameters and efficiency
- Capacity and leakage assessment
- Factors affecting the performance and efficiency
- Types of dryers and their operating principle
- Energy saving opportunities

3. COMPRESSED AIR SYSTEM

Types of air compressors, Comparison of different air compressors, Compressor efficiency, Efficient compressor operation, Compressed air system components, Air dryers, Capacity assessment, Leakage test, Factors affecting the performance and efficiency.

3.1 Introduction

Air compressors account for significant amount of electricity used in Indian industries. Air compressors are used in a variety of industries to supply process requirements, to operate pneumatic tools and equipment, and to meet instrumentation needs. Only 10-30% of energy reaches the point of end-use (Figure 3.1), and balance 70-90% of energy of the power of the prime mover being converted to unusable heat energy and to a lesser extent lost in form of friction, misuse and noise.

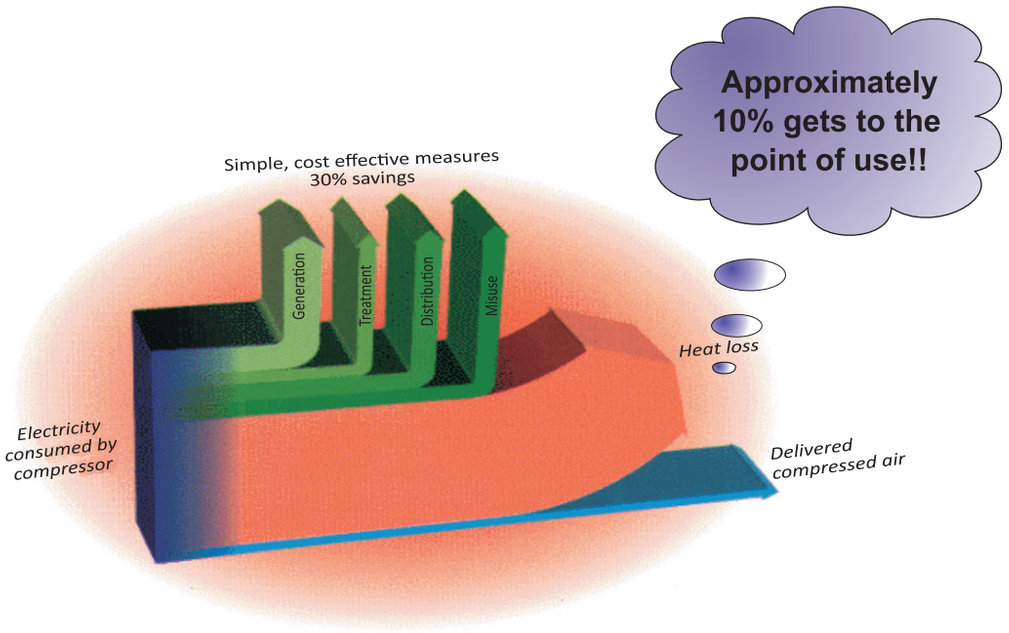

Figure 3.1 Sankey Diagram for Compressed Air System

3.2 Compressor Types

Compressors are broadly classified as: Positive displacement compressor and Dynamic compressor (Figure 3.2).

Positive displacement compressors increase the pressure of the gas by reducing the volume. Positive displacement compressors are further classified as reciprocating and rotary compressors.

Dynamic compressors increase the air velocity, which is then converted to increased pressure at the outlet. Dynamic compressors are basically centrifugal compressors and are further classified as radial and axial flow types.

The flow and pressure requirements of a given application determine the suitability of a particulars type of compressor.

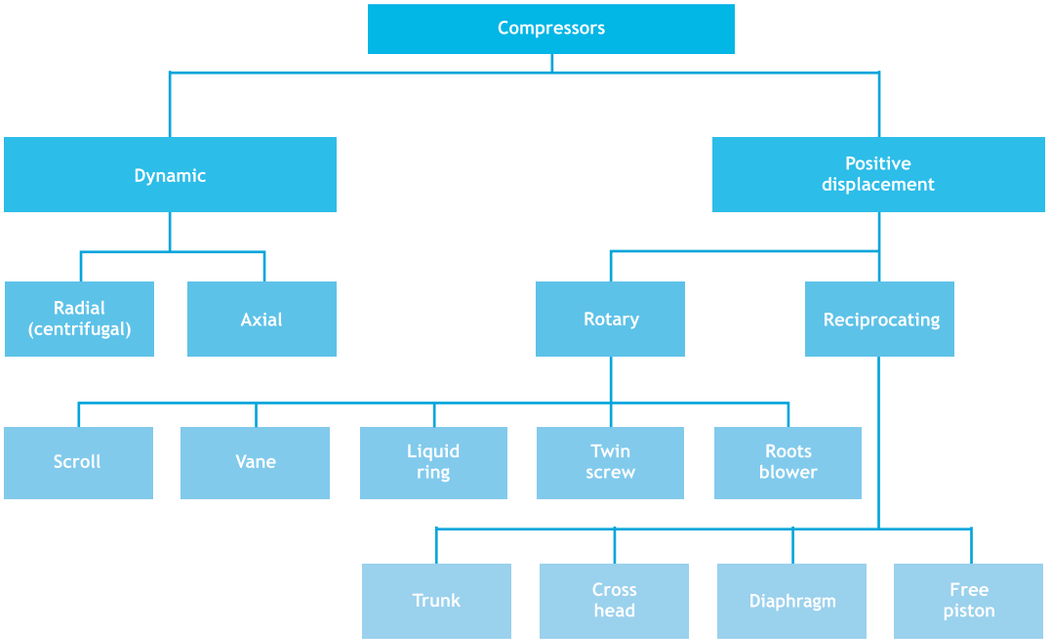

Figure 3.2 Compressor Chart

Positive Displacement Compressors

Reciprocating Compressors

Reciprocating compressors (Figure 3.3) are the most widely used type for air compression. They are characterized by a flow output that remains nearly constant over a range of discharge pressures. Also, the compressor capacity is directly proportional to the speed. The output, however, is a pulsating one.

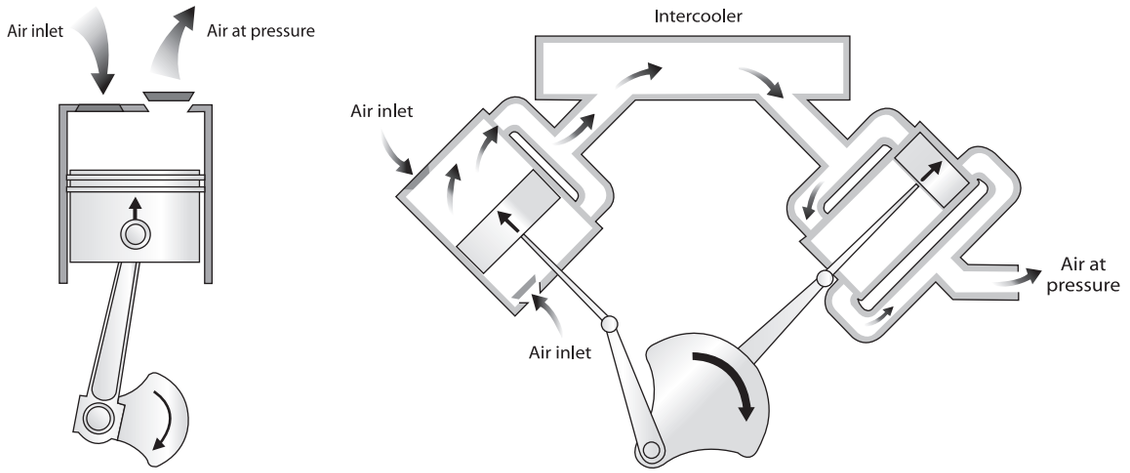

Figure 3.3 Reciprocating Compressor

Reciprocating compressors are available in many configurations, the four most widely used of which are horizontal, vertical, horizontal balance-opposed and tandem. Vertical type reciprocating compressors are used in the capacity range of 50 – 150 cfm. Horizontal balance opposed compressors are used in the capacity range of 200 – 5000 cfm in multi-stage design and up to 10,000 cfm in single stage designs.

Reciprocating compressors are also available in variety of types:

- Lubricated and non-lubricated
- Single or multiple cylinder
- Water or air-cooled.
- Single or multi stage

In the case of lubricated machines, oil has to be separated from the discharge air. Non-lubricated compressors are especially useful for providing air for instrumentation and for processes which require oil free discharge. However non-lubricated machines have higher specific power consumption (kW/cfm) as compared to lubricated types.

Single cylinder machines are generally air-cooled, while multi-cylinder machines are generally water cooled, although multi-stage air-cooled types are available for machines up to 100 kW. Water-cooled systems are more energy efficient than air-cooled systems.

Two stage machines are used for high pressures and are characterized by lower discharge temperature (140 to 160^0C) compared to single-stage machines (205 to 240^0C). In some cases, multi-stage machines may have a lower specific power consumption compared to single stage machines operating over the same total pressure differential. Multi-stage machines generally have higher investment costs, particularly for applications with high discharge pressure (above 7 bar) and low capacities (less than 25 cfm). Multi staging has other benefits, such as reduced pressure differential across cylinders, which reduces the load and stress on compressor components such as valves and piston rings.

Rotary Compressors

Rotary compressors (Figure 3.4) have rotors in place of pistons and give a continuous, pulsation free discharge air. They are directly coupled to the prime mover and require lower starting torque as compared to reciprocating machine. They operate at high speed and generally provide higher throughput than reciprocating compressors. Also they require smaller foundations, vibrate less, and have a lower number of parts - which means less failure rate.

Among rotary compressor, the Roots blower (also called as lobe compressor) and screw compressors are among the most widely used. The roots blower is essentially a low-pressure blower and is limited to a discharge pressure of 1 bar in single-stage design and up to 2.2 bar in two stage design.

The most common rotary air compressor is the single stage helical or spiral lube oil flooded screw air compressor. These compressors consist of two rotors, within a casing where the rotors compress the air internally. There are no valves. These units are basically oil cooled

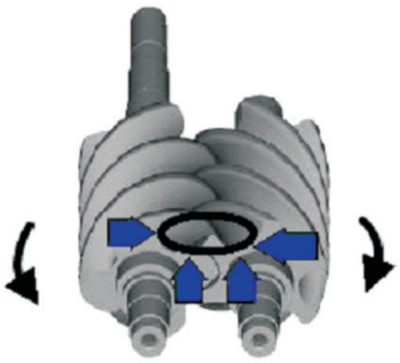

Figure 3.4 Screw Compressors

(with air cooled or water cooled oil coolers) where the oil seals the internal clearances. Since the cooling takes place right inside the compressor, the working parts never experience extreme operating temperatures. The oil has to be separated from discharge air. Because of the simple design and few wearing parts, rotary screw air compressors are easy to maintain, to operate and install.

The oil free rotary screw air compressor uses specially designed air ends to compress air without oil in the compression chamber producing true oil free air. These compressors are available as air-cooled or water cooled types and provide the same flexibility as oil flooded rotary compressors.

There is a wide range of availability in configuration and in pressure and capacity. Dry types deliver oil-free air and are available in sizes up to 20,000 cfm and pressure upto 15 bar. Lubricated types are available in sizes ranging from 100 to 1000 cfm, with discharge pressure up to 10 bar.

Dynamic Compressors

Dynamic compressors (Figure 3.5) are mainly centrifugal compressors and operate on similar principles to centrifugal pump. These compressors have appreciably different characteristics as compared to reciprocating machines. A small change in compression ratio produces a marked change in compressor output and efficiency. Centrifugal machines are better suited for applications requiring very high capacities, typically above 12,000 cfm.

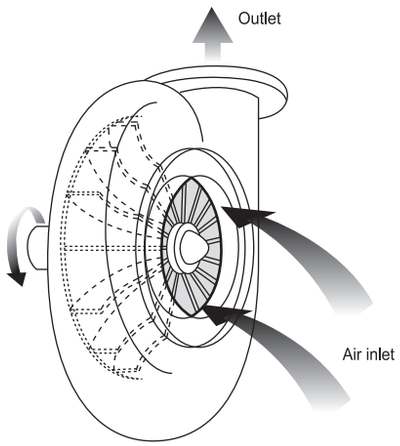

Figure 3.5 Axial Compressor

The centrifugal air compressor is an oil free compressor by design. The oil-lubricated running gear is separated from the air by shaft seals and atmospheric vents. The centrifugal is a continuous duty compressor, with few moving parts, and is particularly suited to high volume applications, especially where oil free air is required.

A single-stage centrifugal machine can provide the same capacity as a multi-stage reciprocating compressor. Machines with either axial or radial flow impellers are available.

The major limitation of a centrifugal compressor is that it operates at peak efficiency at design point only and any deviation from the operating point penalizes efficiency. When selecting centrifugal compressors, close attention should be paid during system design to ensure that at high pressure, with the consequent reduction in flow, the surge point or zone of unstable operation is not reached.

Axial flow compressors are suitable for higher compression ratios and are generally more efficient than radial compressors. Air is allowed to pass through series of alternate rows of rotating blades (attached with rotors) and fixed blades (fixed to casing), having direction of flow parallel to the axis.

The general selection criteria for compressor is given in the Table 3.1

Table 3.1 General Selection Criteria for Compressors				
Type of Compressor	**Capacity (m^3/h)**		**Pressure (bar)**	
	From	**To**	**From**	**To**
Roots blower compressor single stage	100	30000	0.1	1
Reciprocating				
- Single / Two stage	100	12000	0.8	12
- Multi stage	100	12000	12.0	700
Screw				
- Single stage	100	2400	0.8	13
- Two stage	100	2200	0.8	24
Centrifugal	600	300000	0.1	450

Comparison of Different Compressors

The power consumption of various compressors depends on the operating pressure, free air delivery and efficiency etc. The variations in power consumption during unloading/part load operation are more significant and depend on the type of compressor and method of capacity control. The relative efficiencies and part load power consumption of different compressors are given in Table 3.2.

Table 3.2 Comparison of Different Compressors				
Item	**Reciprocating**	**Rotary vane**	**Rotary Screw**	**Centrifugal**
Efficiency at full load	High	Medium-high	High	High
Efficiency at part load	High due to staging	Poor: below 60% of full load	Poor: below 60% of full load	Poor: below 60% of full load
Efficiency at no load (power as % of full load)	High (10-25%)	Medium (30-40%)	High-poor (25-60%)	High-medium (20-30%)

In case of reciprocating machines, the unload power consumption is in the order of 25% of full load power. While in screw compressors, the unload power consumption is marginally higher compared to reciprocating machines.

It is preferable to use screw compressors for constant air requirement. If screw compressors have to be installed for fluctuating loads, it is desirable to have screw compressor with variable speed drive to further optimize unload power consumption.

Some of the plants have adopted the strategy of operating screw compressor at full load for meeting the base-load requirement and reciprocating compressor for fluctuating load to optimize on unload power consumption.

3.3 Compressor Performance

Capacity of a Compressor: Free Air Delivery (FAD)

Free air, as defined by CAGI (Compressed Air & Gas Institute) is air at ATMOSPHERIC conditions at any specific location. Because the barometer and temperature may vary at different localities and at different times, it follows that this term does not mean air under standard conditions.

Measured in CFM (Cubic feet per minute) this is the amount of compressed air converted back to the actual inlet (free air) conditions before it was compressed. In other words, the volume of air, which is drawn in from the atmosphere by the compressor, then compressed and delivered at a specific pressure.

Compressor Efficiency Definitions

Several different measures of compressor efficiency are commonly used: volumetric efficiency, adiabatic efficiency, isothermal efficiency and mechanical efficiency.

Adiabatic and isothermal efficiencies are computed as the isothermal or adiabatic power divided by the actual power consumption. The figure obtained indicates the overall efficiency of compressor and drive motor.

Isothermal Efficiency

$$\text{Isothermal Efficiency} = \frac{\textit{Isothermal Power}}{\text{Actual measured input power}}$$

$$\text{Isothermal power (kW)} = P_1 \times Q_1 \times \log_e r / 36.7$$

r = Pressure ratio, P_2/P_1
P_1 = Absolute intake pressure kg/ cm^2
P_2 = Absolute delivery pressure, kg/cm^2
Q_1 = Free air delivered m^3/hr.

The calculation of isothermal power does not include power needed to overcome friction and generally gives an efficiency that is lower than adiabatic efficiency. The reported value of efficiency is normally the isothermal efficiency. This is an important consideration when selecting compressors based on reported values of efficiency.

Volumetric Efficiency

$$\textit{Volumetric efficiency} = \frac{\textit{Free air delivered}\ (m^3/\min)}{\textit{Compressor displacement}\ (m^3/\min)} x100$$

$$\text{Compressor Displacement} = \frac{\pi}{4} x D^2 x L x S x \chi x n$$

D = Cylinder bore, metre
L = Cylinder stroke, metre
S = Compressor speed rpm
χ = 1 for single acting and
2 for double acting cylinders
n = No. of cylinders

For practical purposes, the most effective guide in comparing compressor efficiencies is the specific power consumption i.e. kW/volume flow rate, for different compressors that would provide identical duty.

3.4 Compressed Air System Components

Compressed air system consist of following major components: Intake air filters, inter-stage coolers, after coolers, air dryers, moisture drain traps, receivers, piping network, filters, regulators and lubricators (see Figure 3.6).

- **Intake Air Filters**: Prevent dust from entering compressor; Dust causes sticking valves, scoured cylinders, excessive wear etc.
- **Inter-stage Coolers**: Reduce the temperature of the air before it enters the next stage to reduce the work of compression and increase efficiency. They are normally water-cooled.
- **After Coolers:** The objective is to remove the moisture in the air by reducing the temperature in a water-cooled heat exchanger.
- **Air-dryers**: The remaining traces of moisture after after-cooler are removed using air dryers because air for instrument and pneumatic equipment has to be relatively free of any moisture. The moisture is removed by using adsorbents like silica gel /activated carbon or refrigerant dryers.
- **Moisture Drain Traps:** Moisture drain traps are used for removal of condensate from the compressed air. These traps resemble steam traps and various types used are manual drain cocks, timer based / automatic drain valves etc.
- **Receivers**: Air receivers are provided as storage and smoothening pulsating air output - reducing pressure variations from the compressor

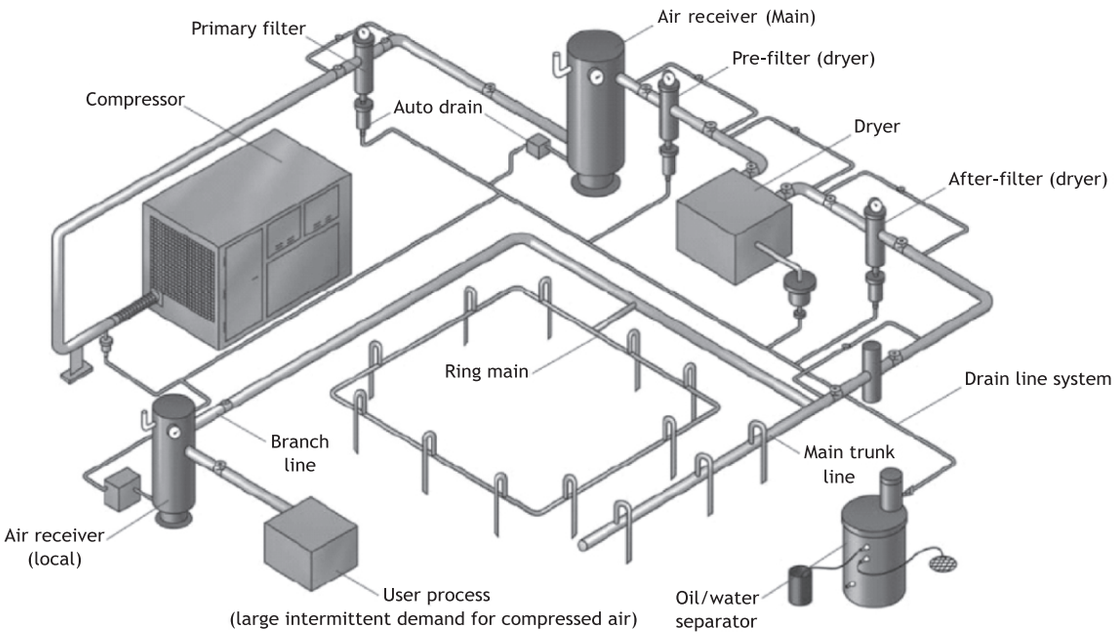

Figure 3.6 A Typical Compressed Air System Components and Network

3.5 Efficient Operation of Compressed Air Systems

Location of Compressors

The location of air compressors and the quality of air drawn by the compressors will have a significant influence on the amount of energy consumed. Compressor performance as a breathing machine improves with cool, clean, dry air at intake.

Cool air intake

As a thumb rule, *"Every 4^{0}C drop in inlet air temperature results in lower energy consumption by 1 % to achieve equivalent output"*. Hence, cool air intake leads to a more efficient compression (see Table 3.3).

Table 3.3 Effect of Intake Air Temperature on Power Consumption

Inlet Temperature (^{0}C)	Relative Air Delivery (%)	Power Saved (%)
10.0	102.0	+ 1.4
15.5	100.0	Nil
21.1	98.1	- 1.3
26.6	96.3	- 2.5
32.2	94.1	- 4.0
37.7	92.8	- 5.0
43.3	91.2	- 5.8

Since heat is generated during operation of compressors, temperature of air inside the compressor room will be higher than the ambient temperature. Hence, it is preferable to draw cool ambient air from outside. While extending air intake from the outside of room, care should be taken to minimize excess pressure drop in the suction line, by selecting a large diameter duct with minimum number of bends.

Dust Free Air Intake

Dust in the suction air causes excessive wear of moving parts and results in malfunctioning of the valves due to abrasion. Suitable air filters should be provided at the suction side. Air filters should have high dust separation capacity, low-pressure drops and robust design to avoid frequent cleaning and replacement. See Table 3.4 for effect of pressure drop across air filter on power consumption.

Air filters should be selected based on the compressor type and installed as close to the compressor as possible. As a thumb rule "For every 250 mm WC pressure drop increase across at the suction path due to choked filters etc, the compressor power consumption increases by about 2 percent for the same output"

Hence, it is advisable to clean inlet air filters at regular intervals to minimize pressure drops. Manometers or differential pressure gauges across filters may be provided for monitoring pressure drops so as to plan filter-cleaning schedules.

Table 3.4 Effect of Pressure Drop Across Air Inlet Filter on Power Consumption	
Pressure Drop Across air filter (mmWC)	**Increase in Power Consumption (%)**
0	0
200	1.6
400	3.2
600	4.7
800	7.0

Dry Air Intake

Atmospheric air always contains some amount of water vapour, depending on the relative humidity, being high in wet weather. The moisture level will also be high if air is drawn from a damp area - for example locating compressor close to cooling tower, or dryer exhaust is to be avoided (see Table 3.5).

Table 3.5 Moisture in Ambient Air at Various Humidity Levels	
% Relative Humidity	**kg of water vapour per hour for every 1000 m^3/min. of air at 30 ^{0}C**
50	27.60
80	45.00
100	68.22

The moisture-carrying capacity of air increases with a rise in temperature and decreases with increase in pressure.

Elevation

The altitude of a place has a direct impact on the volumetric efficiency of the compressor. The effect of altitude on volumetric efficiency is given in the Table 3.6.

It is evident that compressors located at higher altitudes consume more power to achieve a particular delivery pressure than those at sea level, as the compression ratio is higher.

Cooling Water Circuit

Most of the industrial compressors are water-cooled, wherein the heat of compression is removed by circulating cold water to cylinder heads, inter-coolers and after-coolers. The resulting warm water is cooled in a cooling tower and circulated back to compressors. The compressed air system performance depends upon the effectiveness of inter-coolers, after coolers, which in turn are dependent on cooling water flow and temperature.

Further, inadequate cooling water treatment can lead to increase, for example, in total dissolved solids (TDS), which in turn can lead to scale formation in heat exchangers. The scales, not only act as insulators reducing the heat transfer, but also increase the pressure drop in the cooling water pumping system.

Table 3.6 Effect of Altitude on Volumetric Efficiency			
Altitude Meters	**Barometric Pressure milli bar***	**Percentage Relative Volumetric Efficiency Compared with Sea Level**	
		At 4 bar	**At 7 bar**
Sea level	1013	100.0	100.0
500	945	98.7	97.7
1000	894	97.0	95.2
1500	840	95.5	92.7
2000	789	93.9	90.0
2500	737	92.1	87.0

*** 1 milli bar = 1.01972 x 10^{-3} kg/cm^2**

Use of treated water or purging a portion of cooling water (blow down) periodically can maintain TDS levels within acceptable limits. It is better to maintain the water pH by addition of chemicals, and avoid microbial growth by addition of fungicides and algaecides.

Efficacy of Inter and After Coolers

Efficacy is an indicator of heat exchange performance- how well intercoolers and after coolers are performing. Inter-coolers are provided between successive stages of a multi-stage compressor to reduce the work of compression (power requirements) - by reducing the specific volume through cooling the air - apart from moisture separation.

Ideally, the temperature of the inlet air at each stage of a multi-stage machine should be the same as it was at the first stage. This is referred to as "perfect cooling" or isothermal compression. The cooling may be imperfect due to reasons described in earlier sections. Hence in actual practice, the inlet air temperatures at subsequent stages are higher than the normal levels resulting in higher power consumption, as a larger volume is handled for the same duty (See Table 3.7).

Table 3.7 Effect of Inter-stage Cooling on Specific Power Consumption of a Reciprocating Compressor -Illustration			
Details	**Imperfect Cooling**	**Perfect Cooling (Base Value)**	**Chilled Water Cooling**
First Stage inlet temperature ^{0}C	21.1	21.1	21.1
Second Stage inlet temperature ^{0}C	26.6	21.1	15.5
Capacity (Nm^3/min)	15.5	15.6	15.7
Shaft Power (kW)	76.3	75.3	74.2
Specific energy consumption (kW/ Nm^3/min)	4.9	4.8	4.7
Percent Change	+ 2.1	Reference	- 2.1

It can be seen from the Table 3.7 that *an increase of 5.5^{0}C in the inlet air temperature to the second stage results in a 2 % increase in the specific energy consumption*. Use of water at lower temperature reduces specific power consumption. However, very low cooling water temperature could result in condensation of moisture in the air, which if not removed would lead to cylinder damage.

Similarly, inadequate cooling in after-coolers (due to fouling, scaling etc.), allow warm, humid air into the receiver, which causes more condensation in air receivers and distribution lines, which in consequence, leads to increased corrosion, pressure drops and leakages in piping and end-use equipment. Periodic cleaning and ensuring adequate flow at proper temperature of both inter coolers and after coolers are therefore necessary for sustaining desired performance. Typical cooling water requirement is given in Table 3.8.

Table 3.8 Typical Cooling Water Requirements

Compressor Type	Minimum quantity of Cooling Water required (in litres per minute) for 2.85 m^3/min. FAD at 7 bar
Single-stage	3.8
Two-stage	7.6
Single-stage with after-cooler	15.1
Two-stage with after-cooler	18.9

Pressure Settings

Compressor operates between pressure ranges called as loading (cut-in) and unloading (cut-out) pressures. For example, a compressor operating between pressure setting of 6 - 7 kg/cm^2 means that the compressor unloads at 7 kg/cm^2 and loads at 6 kg/cm^2. Loading and unloading is done using a pressure switch.

For the same capacity, a compressor consumes more power at higher pressures. They should not be operated above their optimum operating pressures as this not only wastes energy, but also leads to excessive wear, leading to further energy wastage. The volumetric efficiency of a compressor is also less at higher delivery pressures.

Reducing Delivery Pressure:

The possibility of lowering (optimising) the delivery pressure settings should be explored by careful study of pressure requirements of various equipments, and the pressure drop in the line between the compressed air generation and utilization points. Typical power savings through pressure reduction is shown in Table 3.9.

Table 3.9 Typical Power Savings Through Pressure Reduction

Pressure Reduction		Power Savings (%)		
From (bar)	To (bar)	Single-stage Water-cooled	Two-stage Water-cooled	Two-stage Air-cooled
6.8	6.1	4	4	2.6
6.8	5.5	9	11	6.5

The pressure switches must be adjusted such that the compressor cuts-in and cuts-out at optimum levels.

A reduction in the delivery pressure by 1 bar in a compressor would reduce the power consumption by 6 – 10 % . For example, reduction of delivery pressure by 1 kg/cm^2 (from 8 kg/cm^2 to 7 kg/cm^2) would result in 9% input power savings.

Compressor modulation by Optimum Pressure Settings:

Very often in an industry, different types, capacities and makes of compressors are connected to a common distribution network. In such situations, proper selection of a right combination of compressors and optimal modulation of different compressors can conserve energy.

When more than one compressor feeds a common header, compressors have to be operated in such a way that the cost of compressed air generation is minimal.

- If all compressors are similar, the pressure setting can be adjusted such that only one compressor handles the load variation, whereas the others operate more or less at full load.
- If compressors are of different sizes, the pressure switch should be set such that only the smallest compressor is allowed to modulate (vary in flow rate).
- If different types of compressors are operated together, unload power consumptions are significant. The compressor with lowest no load power must be modulated.
- In general, the compressor with lower part load power consumption should be modulated.
- Compressors can be graded according to their specific energy consumption, at different pressures and energy efficient ones must be made to meet most of the demand (see Table 3.10)

Table 3.10 Typical Specific Power Consumption of Reciprocating Compressors (based on motor input)

Pressure bar	No. of Stages	Specific Power kW/170 m^3/hour (kW / 100 cfm)
1	1	6.29
2	1	9.64
3	1	13.04
4	2	14.57
7	2	18.34
8	2	19.16
10	2	21.74
15	2	26.22

EXAMPLE

Compressor modulation

Assessing compressed air system study for a plant section gave following results. Comment on the results?

- Compressors on line A, B, C, D, E (all reciprocating type)
- Trial observation Summary

Compressor Reference	Measured Capacity CMM (@ 7 kg/ cm^2)	'On' Load kW	'Unload' kW	Load Time min.	Unload Time min.
A	13.17	115.30	42.3	Full time*	Nil
B	12.32	117.20	51.8	Full time*	Nil
C	13.14	108.30	43.3	Full time*	Nil
D	12.75	104.30	29.8	Full time*	Nil
E	13.65	109.30	39.3	5.88	39.12

* Compressors running in load conditions and not getting unloaded during normal operations.

Comments:

For a cycle time of 45 minutes (39.12 + 5.88)

i) Compressed air generated in m^3

= 45 (13.17) + 45 (12.32) + 45 (13.14) + 45 (12.75) + 5.88 (13.65)

= 2392.36 m^3

ii) Power consumption kWh

= 45/60 (115.3) + 45/60 (117.20) + 45 / 60 (108.3) + 45/60 (104.3) + 5.88/60 (109.30) + 39.12 / 60 (39.3)

= 370.21 kWh / 45 minutes

iii) Compressed air generation actual capacity on line in m^3

= 45 [13.17 + 12.32 + 13.14 + 12.75 + 13.65] = 2926.35 m^3

a) The consumption rate of the section connected

= 2392.36 / 45 = 53.16 m^3/minute

b) Compressor air drawl as a % of capacity on line is

= [2392.36 / 2926.35] × 100 = 81.75 %

c) Specific power consumption = 370.21 / 2392.36 = 0.155 kW/m^3

d) Idle power consumption due to unload operation = 25.62 [=39.12 x 39.3 / 60] kWh in every 45 minutes cycle i.e., 34.16 [=25.62 x 60 / 45] kWh every hour.

e) It would be favourable in short term and energy efficient to keep the compressor 'D' in cycling mode on account of lower un-load losses and hence capacity. Speed of the compressor can also be reduced by reducing motor pulley size.

f) A suitable smaller capacity compressor can be planned to replace the compressor with highest unload losses.

g) An investigation is called for, as to why such a large variation of unload power drawn, exists although all compressors have almost the same rated capacity.

Segregating low and high pressure air requirements

If the low-pressure air requirement is considerable, it is advisable to generate low pressure and high-pressure air separately, and feed to the respective sections instead of reducing the pressure through pressure reducing valves, which invariably waste energy.

Minimum pressure drop in air lines

Pressure drop in air lines is depending upon the quantity of airflow, diameter of the pipeline, pipe length and pipe geometry i.e. the bends in the pipe lines.-Excess pressure drop due to inadequate pipe sizing, choked filter elements, improperly sized couplings and hoses represent energy wastage

The pipelines should be with minimum number of Joints, bends and fittings. Further to minimize the joints, it should be ensured that joints are welded instead of flexible or screwed joints. This facilitates minimizing the leakages and pressure drop.

Typical acceptable pressure drop in industrial practice is 0.3 bar in mains header at the farthest point and 0.5 bar in distribution system.

The Table 3.11 illustrates the energy wastage, if the pipes are of smaller diameter.

Table 3.11 Typical Energy Wastage due to Smaller Pipe Diameter for 170 m^3/h (100 cfm) Flow

Pipe Nominal Bore (mm)	Pressure drop (bar) per 100 meters	Equivalent power losses (kW)
40	1.80	9.5
50	0.65	3.4
65	0.22	1.2
80	0.04	0.2
100	0.02	0.1

Equivalent lengths of fittings

Not only piping, but also fitting are a source of pressure *losses. Typical pressure losses for various fitting are given in Table 3.12.

Table 3.12 Resistance of Pipe Fittings in Equivalent Lengths (in metres)

Type of Fitting	Nominal Pipe Size in mm									
	15	20	25	32	40	50	65	80	100	125
Gate Valve	0.11	0.14	0.18	0.27	0.32	0.40	0.49	0.64	0.91	1.20
Tee 90^0 long bend	0.15	0.18	0.24	0.38	0.46	0.61	0.76	0.91	1.20	1.52
Elbow	0.26	0.37	0.49	0.67	0.76	1.07	1.37	1.83	2.44	3.20
Return bend	0.46	0.61	0.76	1.07	1.20	1.68	1.98	2.60	3.66	4.88
Outlet of tee globe valve	0.76	1.07	1.37	1.98	2.44	3.36	3.96	5.18	7.32	9.45

Blowers in place of Compressed Air System

Since the compressed air system is already available, plant engineer may be tempted to use compressed air to provide air for low-pressure applications such as agitation, pneumatic conveying or combustion air. Using a blower that is designed for lower pressure operation will cost only a fraction of compressed air generation energy and cost.

Capacity Control of Compressors

In many installations, the use of air is intermittent. Therefore, some means of controlling the output flow from the compressor is necessary. The type of capacity control chosen has a direct impact on the compressor power consumption. Some control schemes commonly used are discussed below:

Automatic On / Off Control:

Automatic On /Off control, as its name implies, starts or stops the compressor by means of a pressure activated switch as the air demand varies. This is a very efficient method of controlling the capacity of compressor, where the motor idle-running losses are eliminated, as it completely switches off the motor when the set pressure is reached. This control is suitable for small compressors.

Load and Unload

This is a two-step control where compressor is loaded when there is air demand and unloaded when there is no air demand. During unloading, a positive displacement compressor may consume up to 30 % of the full load power, depending upon the type, configuration, operation and maintenance practices.

Multi-step Control

Large capacity reciprocating compressors are usually equipped with a multi-step control. In this type of control, unloading is accomplished in a series of steps, (0%, 25 %, 50 %, 75 % & 100 %) varying from full load down to no-load (see Table 3.13).

Table 3.13 Power Consumption of a Typical Reciprocating Compressor at Various Loads

Load %	Power Consumption as % of full load Power
100	100
75	80
50	60
25	40
0	20-25

Throttling Control

The capacity of centrifugal compressors can be controlled using variable inlet guide vanes. However, another efficient way to match compressor output to meet varying load requirements is by speed control (see Table 3.14).

Table 3.14 Typical Part Load Air Compression :Power Input for Speed and Vane Control of Centrifugal Compressors		
System Volume, %	**Power Input (%) Speed Control**	**Power Input (%) Vane Control**
111	120	-
100	100	100
80	76	81
60	59	64
40	55	50
20	51	46
0	47	43

At low volumetric flow (below 40 %), vane control may result in lower power input compared to speed control due to low efficiency of the speed control system. For loads more than 40 %, speed control is recommended.

Avoiding Misuse of Compressed Air:

Misuse of compressed air for purposes like body cleaning, liquid agitation, floor cleaning, drying, equipment cooling and other similar uses must be discouraged. Wherever possible, low-pressure air from a blower should be substituted for compressed air, for example secondary air for combustion in a boiler / furnace.

The following Table 3.15 gives an idea of savings by stopping use of compressed air by choosing alternative methods to perform the same task.

- Electric motors can serve more efficiently than air-driven rotary devices, wherever applicable. The Table gives the comparison of pneumatic grinders and electrical grinders.

Table 3.15 Typical Power Requirements for Pneumatic and Electrical Tools				
Tool	**Wheel dia, mm**	**Speed, rpm**	**Air Consumption, m^3/h**	**Power, kW**
Pneumatic angle grinder	150	6000	102 m^3/h at 6 bar	10.2
Electric angle grinder	150	5700-8600	N.A.	1.95 – 2.90
Pneumatic jet grinder	35	30000	32.3 m^3/h at 6 bar	3.59
Electric straight grinder	25	22900 - 30500	N.A.	0.18

It may be noted that in some areas use of electric tools are not permitted due to safety constraints, especially places where inflammable vapours are present in the environment. It should always be remembered that safety consideration always overrides energy conservation.

- In place of pneumatic hoists, electric hoists can be used.
- Material conveying applications by blower systems can be replaced preferably by a combination of belt / screw conveyers and bucket elevators. In a paper manufacturing facility, compressed air was used for conveying wood chips. The equivalent power consumption was 77 kW. This

method of conveying was replaced by blower system consuming only 7 kW, a saving of 70 kW. This has also been widely applied in cement industry where pneumatic conveying has been replaced by bucket and screw conveyor resulting in significant energy reduction.

How air amplifiers work?

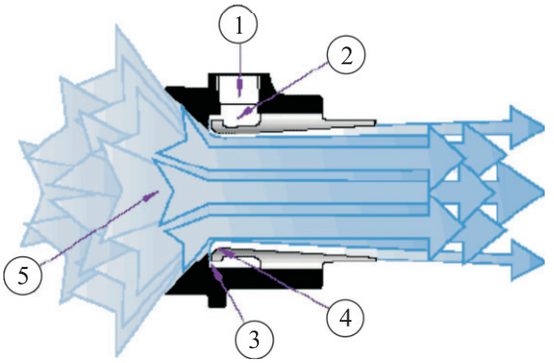

Figure 3.7

Compressed air flows through the inlet (1) into an annular chamber (2). It is then throttled through a small ring nozzle (3) at high velocity. This primary air stream adheres to the coanda profile (4), which directs it toward the outlet. A low pressure area is created at the center (5) inducing a high volume flow of surrounding air into the primary air stream. The combined flow of primary and surrounding air exhausts from the Air Amplifier in a high volume, high velocity flow.

- When moving air really is required for an application, often sources other than compressed air can do the job. For applications like blowing of components, use of compressed *air amplifiers* (see Figure 3.7*)*, blowers or gravity-based systems may be possible. Brushes can sweep away debris from work in progress as effectively as high-pressure air. Blowers can also be used for this purpose. Many applications do not require clean, dry, high-pressure and expensive 6 bar or 7 bar compressed air rather, only moving air is needed to blow away debris, provide cooling, or other functions. In these cases, local air fans or blowers may satisfy the need for moving air much economically. If a ¼" hose pipe is kept open at a 7 bar compressed air line for cleaning for at least 1000 hours / annum, it can cost about Rs. 1.0 lakhs / annum. If absolutely necessary, compressed air should be used only with blow guns to keep the air pressure below 2 bar.

- For applications, where compressed air is indispensable for cleaning internal crevices of machines etc., installation of a separate cleaning air header with a main isolation valve may be considered. The main valve should be opened only for a few, well-defined time periods during the whole day; no connections for cleaning should be provided from process or equipment air lines.

- Replacement of pneumatically operated air cylinders by hydraulic power packs can be considered.

- Vacuum systems are much more efficient than expensive venturi methods, which use expensive compressed air rushing past an orifice to create a vacuum.

- Mechanical stirrers, conveyers, and low-pressure air will mix materials far more economically than high-pressure compressed air.

Avoiding Air Leaks and Energy Wastage:

The major opportunity to save energy is in the prevention of leaks in the compressed air system. Leaks frequently occur at air receivers, relief valves, pipe and hose joints, shut off valves, quick release couplings, tools and equipment. In most cases, they are due to poor maintenance and sometimes, improper installations etc.

Air leakages through Different Size Orifices

The Table 3.16 gives the amount of free air wasted for different nozzles sizes and pressure.

Table 3.16 Discharge of Air (m^3/minute) through Orifice (Orifice Constant C_d – 1.0)

Gauge Pressure Bar	0.5 mm	1 mm	2 mm	3 mm	5 mm	10 mm	12.5 mm
0.5	0.06	0.22	0.92	2.1	5.7	22.8	35.5
1.0	0.08	0.33	1.33	3.0	8.4	33.6	52.5
2.5	0.14	0.58	2.33	5.5	14.6	58.6	91.4
5.0	0.25	0.97	3.92	8.8	24.4	97.5	152.0
7.0	0.33	1.31	5.19	11.6	32.5	129.0	202.0

Cost of Compressed Air Leakage:

It may be seen from Table 3.17 that any expenditure on stopping leaks would be paid back through energy saving.

Table 3.17 Cost of Air Leakage

Orifice Size mm	kW Wasted	* Cost of air leakage (Rs/Year)
0.8	0.2	8000
1.6	0.8	32000
3.1	3.0	120000
6.4	12.0	480000

* based on Rs. 5 / kWh; 8000 operating hours; air at 7.0 bar

Steps in simple shop-floor method for leak quantification

- Shut off compressed air operated equipments (or conduct test when no equipment is using compressed air).
- Run the compressor to charge the system to set pressure of operation
- Note the sub-sequent time taken for 'load' and 'unload' cycles of the compressors. For accuracy, take ON & OFF times for 8 – 10 cycles continuously. Then calculate total 'ON' Time (T) and Total 'OFF' time (t).
- The system leakage is calculated as:

$$\%\ leakage = \frac{T}{(T+t)}\ x\ 100$$

$$\text{(or)}\ \ System\ leakage\ quantity\ (m^3/\min), q = \frac{T}{(T+t)} x Q$$

Where,

Q = Compressor capacity (m^3/min)
T = Time on load in minutes
t = Time on unload in minutes

EXAMPLE

In the leakage test in a process industry, following results were observed

Compressor capacity (m^3/minute)	=	35
Cut in pressure, kg/cm^2(g)	=	6.8
Cut out pressure, kg/cm^2(g)	=	7.5
Load kW drawn	=	188 kW
Unload kW drawn	=	54 kW
Average 'Load' time, T	=	1.5 minutes
Average 'Unload' time, t	=	10.5 minutes

Comment on leakage quantity and avoidable loss of power due to air leakages.

a) Leakage quantity (m^3/minute), q = $\frac{(1.5)}{(1.5)+(10.5)} \times 35$

= 4.375 m^3/min

b) Leakage quantity per day, (m^3/day) = 4.375 x 24 x 60 = 6300 m^3/day

c) Specific power for compressed air generation = 188 kW /(35 x 60) m^3/hr

= 0.0895 kWh/m^3

d) Energy lost due to leakage/day = 0.0895 x 6300 = 564 kWh

Leakage Detection by Ultrasonic Leak Detector

Leakage tests are conducted by a Leak Detector having a sensing probe, which senses when there are leakages in compressed air systems at high temperatures-beneath insulated coverings, pipelines, manifolds etc.

The leak is detected by ultrasonic vibration. Leak testing is done by observing and locating sources of ultrasonic vibrations created by turbulent flow of gases passing through leaks in pressurized or evacuated systems.

Line Moisture Separator and Traps

Although, in an ideal system, all cooling and condensing of air should be carried out before the air leaves the receiver, this is not very often achieved in practice. The amount of condensation, which takes place in the lines, depends on the efficiency of moisture extraction before the air leaves the receiver and the temperature in the mains itself. In general, the air main should be given a fall of not less than 1 m in 100 m in the direction of air flow, and the distance between drainage points should not exceed 30m.

Drainage points should be provided using equal tees, as it assists in the separation of water. Whenever a branch line is taken off from the mains it should leave at the top so that any water in the main does not fall straight into the plant equipment. Further, the bottom of the falling pipe should also be drained.

Compressed Air Filter

Although, some water, oil and dirt are removed by the separators and traps in the mains, still some are always left, which are carried over along with compressed air. Moreover, pipe systems accumulate scale and other foreign matters, such as small pieces of gasket material, jointing compounds etc. Burnt compressor oil may also be carried over in pipe work, and this, with other contaminants, forms a gummy substance. To remove these, all of which are liable to have harmful effects on pneumatic equipment, the air should be filtered as near as possible to the point of use. Water and oil collected in the filter sump must be drained off; because if the level is allowed to build up, it is forced through the filter element into the very system for which it is designed to protect.

Regulators

In many instances, pneumatic operations are to be carried out at a lower pressure than that of the main supply. For these applications, pressure regulators are required to reduce the pressure to the required value and also to ensure that it remains reasonably constant at the usage point.

Lubricators

Where air is used to drive prime movers, cylinders and valves, they should be fitted with a lubricator. Essentially, a lubricator is a reservoir of oil and has been designed so that when air is flowing, a metered amount of oil is fed in mist form into the air stream. This oil is carried with the motive air, to the point of use to lubricate all moving parts. All lubricators require a certain minimum rate of airflow to induce oil into their stream. It is advisable to install filters, regulators and lubricators as close as possible to the equipment being served.

Air Dryers

The atmospheric air has certain amount of moisture. The moisture holding capacity of air depends on the ambient temperature. Higher the temperature more is the moisture holding capacity of air in the form of water vapour and vice versa. Saturated air at a given temperature is the air that contains the maximum amount of water in the form of water vapour. Any excess water vapour will be condensed in the form of water.

About 60 to 75% of moisture in compressed air is removed at the after cooler. This is sufficient for many plant air applications such as cleaning, atomisation etc. As compressed air leaves the after cooler and passes through the compressed air lines, the temperature of the compressed air further reduces. The remaining water vapour in the air starts condensing.

The effects of water particles in the compressed air are given below.

- The water particles travel at the same velocity of compressed air and damages the pneumatic valves or instruments at the user ends by erosion.
- Corrosion in the distribution pipe work.
- Impaired finishing processes particularly in paint spraying, sheet cleaning etc.

Hence, the water vapour from the compressed air has to be removed, for applications such as instrumentation and pneumatics. This can be achieved by passing the compressed air through the air dryers.

The extent of drying compressed air is expressed by the term "Atmospheric Dew Point" which is the temperature at which moisture present in the air starts condensing at atmospheric pressure. Lower the dew point, more dry is the air. Air at -40^{0}C atmospheric dew point means no moisture would condense unless temperature of the air is reduced to less than – 40^{0}C, at atmospheric pressure.

The moisture content corresponding to various atmospheric dew points is shown in the table 3.18 below:

Table 3.18 Moisture Content and Atmospheric Dew Point

Atmospheric Dew Point (^{o}C)	0	-5	-10	-20	-30	-40	-60	-80
Moisture Content (PPM)	3800	2500	1600	685	234	80	6.5	0.3

The performance of a dryer is quoted in terms of 'pressure dew point' and there is a relation between both dew points.

Increasing the pressure of a gas increases the dew point temperature of the gas. For example, air at atmospheric pressure of 1013.3 mbar has a dew point temperature of -10 °C. Corresponding partial pressure of water vapour is 2.8 mbar. If this air is compressed and the total pressure is doubled to 2026.6 mbar, then according to Dalton's law, the partial pressure of water vapour is also doubled to the value of 5.6 mbar. The dew point temperature corresponding to 5.6 mbar is approximately -1 °C, so it is clear that increasing the pressure of the air also increases the dew point temperature of the air.

Most commonly used dryers in the industry are:

1. Refrigerant type and
2. Adsorption type - can be of the following type:

   (a) Blower reactivated type

   (b) Heatless purge type

   (c) Heat of compression (HOC) type

The principle of operation of different dryers are explained as follows.

Refrigerant Dyers

The system (Figure 3.8) is straight mechanical refrigeration in which the dew point of air is reduced by chilling. Additionally, a second heat exchanger is introduced wherein the outgoing cold air pre-cools the incoming compressed air. The compressed air leaves the system fully dry. The achievable dew point (atmospheric) in this kind of refrigerant dryer is -20°C.

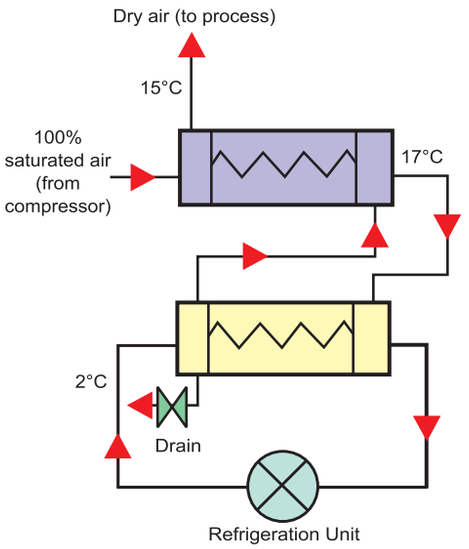

Figure 3.8 Refrigeration Dryer

Features:

- Refrigeration drying is the most economical process of approximately 90 % of all applications.
- Almost 100 % of all solid particles and water droplets larger than 3 µm are separated.
- The pressure loss from the dryer is approximately 0.2 bar.

Adsorption Drying

Drying compressed air by adsorption is a purely physical process. The moisture is bound to the drying agent by force of adhesion (unbalanced molecular attraction). The moisture stays on the inner and outer surfaces of the adsorption material without a chemical reaction taking place. The adsorption material has an open porous structure and a large inner surface. The most common adsorption materials used for compressed air drying are activated alumina and silica gel.

Operating principle

During the drying process the moist compressed air flows through an adsorption tank. The moisture is bound, which dries the compressed air. This process generates heat. The adsorption material must be regenerated when the adhesive forces are balanced by water deposits. This means that the water must be removed from the adsorption material. For this reason there must be second parallel drying tank to maintain continual operation. The active tank A (Figure 3.9) dries the compressed air, while the inactive tank B regenerates without pressure. Based on the method of regeneration the adsorption dryers can be classified as follows.

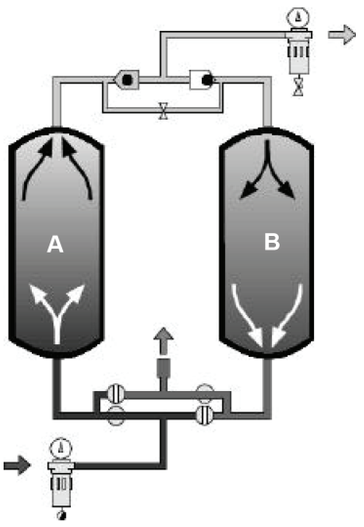

Figure 3.9 Adsorption Dryer

(i) Blower Reactivated Type Dryer

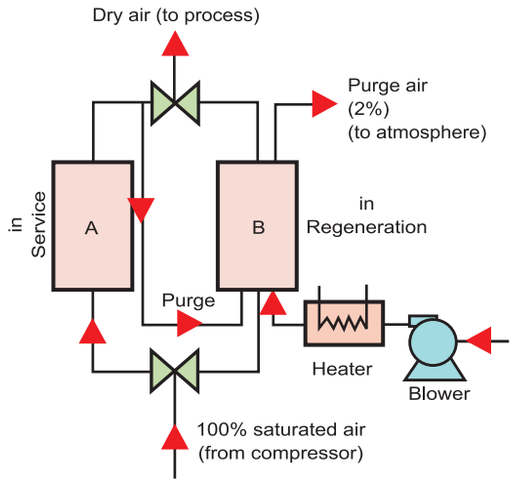

Figure 3.10 Blower Reactivated Type Dryer

A blower and external heater is used to achieve the regenerating temperature. The operating cost is higher because of the heater (electrical or steam) and also because there is a purge loss of about 1-2% of dry compressed air. The vessel which is regenerated is purged with dry air before it is taken into service. These dryers are normally used for capacities higher than 250 cfm. The schematic of the dryer is shown in Figure 3.10. Vessel `A' is in service and vessel `B' is in heating mode. Cooling is done after which vessel `B' is taken into service and vessel `A' goes into regeneration mode.

(ii) Heatless Purge Type Dryer

The operation is very similar to the blower reactivated type, only difference being that no heating of the dessicant is done. Pure dry compressed air is used for purging through the saturated dessicant. These are used for capacities less than 250 cfm. The operating cost is very high due to purge losses of about 12-15%. The schematic of the dryer is shown in Figure 3.11.

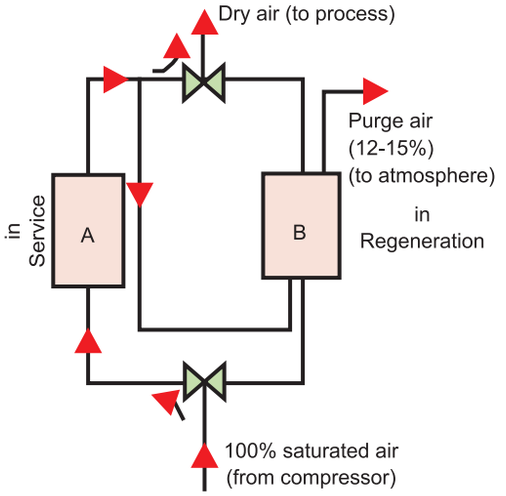

Figure 3.11 Heatless Purge Type Dryer

(iii) Heat of Compression (HOC) Dryer

This is a break-through in drying technology where the operating cost is zero or very minimal. HOC dryers are available from 400 cfm to 5000 cfm capacity. Compressed air, directly from compressor discharge (before after cooler), which is at a temperature of 135 ^{o}C (in the case of reciprocating compressor) is used to regenerate the dessicant. There are no electrical heaters and no purging loss. This makes the dryer very attractive in terms of operating cost.

The dryer consists of two vessels - `A' and `B'. Vessel `A' will be in service for 4 hours. Meanwhile vessel `B' is reactivated which consists of heating for 2 1/2 hrs and cooling for 1 1/2 hrs. After this, vessel `B' is taken into service and vessel `A' is reactivated. The regeneration cycle consisting of heating and cooling cycle as explained as follows

a. Vessel `A' in service, vessel `B' in heating

Air from compressor enters 4-way valves V2 and V1 (Figure 3.12 (a)) and directly to vessel `B' so as to start the heating process. From vessel `B' the air through valve V3 and V2 enters after cooler AC1, where it loses some of the moisture. Through V3 again air enters vessel `A' where moisture is adsorbed by the dessicant and finally leaves through V1 to an After-cooler AC2 where it is cooled to about 35-40 °C. After getting filtered in the After-filter, air goes to process which is dry to an atmospheric dew point of -40 °C. The heating cycle is normally for 2 1/2 hours duration.:

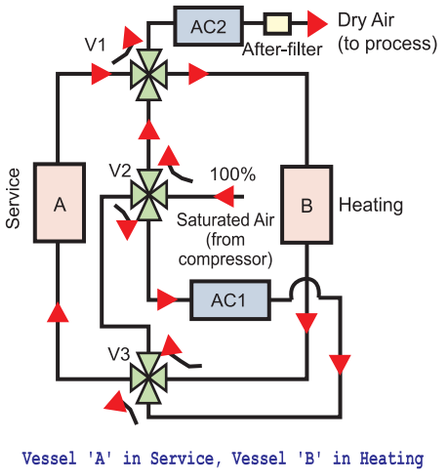

Figure 3.12 (a) Heat of Compression Dryer

b. Vessel `A' in service, vessel `B' in cooling

Air from compressor passes through V2 (Figure 3.12 (b)), gets cooled in AC1 and enters vessel `B' through V3. After cooling the dessicant in vessel `B' air passes through 4-way valves V1, V2 and V3 and enters vessel `A', which is in service. The air on getting dry, enters After-cooler AC2 via V1, gets cooled to about 35-40 °C. This air dryed to an atmospheric dew point of -40 °C is now ready for use. The cooling cycle is normally for about 11/2 hour duration.

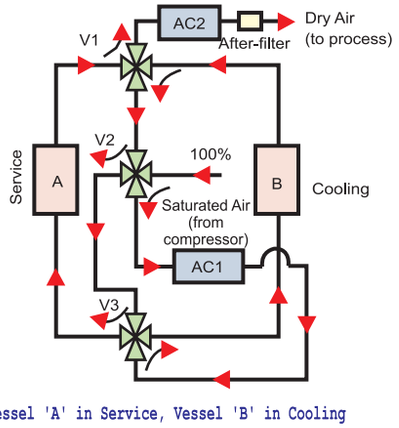

Figure 3.12 (b) Heat of Compression Dryer

Membrane Dryers

Membrane dryers use the process of selective permeation of the gas components in the air. The dryer is a cylinder which houses thousands of tiny hollow polymer fibers with an inner coating. These fibers have selective permeation for the removal of water vapor. As filtered, wet compressed air enters the cylinder, the membrane coating allows water vapor to permeate the membrane wall and collect between the fibers, while the dry air continues through the fibers in the cylinder at almost the same pressure as the incoming wet air. The permeated water is vented to the atmosphere outside of the cylinder.

The permeation or separation is caused by the difference in the partial pressure of a gas between the inside and the outside of the hollow fiber.

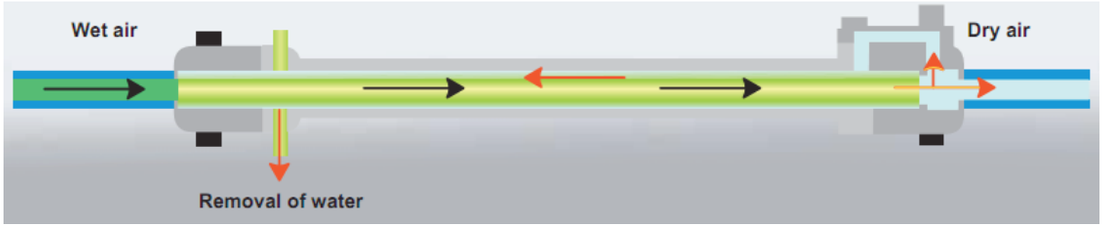

Figure 3.13 Working Principle of Membrane Dryer

Membrane dryers (Figure 3.13) are simple to operate, silent while operating, have no moving parts, low power consumption and minimal service requirements (mainly filters upstream of the dryer).

Refer Table 3.19 for typical dew point and power consumption data for dryers.

Table 3.19 Typical Dew Point and Power Consumption Data for Dryers

Type of Dryer	Dew Point ^{0}C	First Cost	Operating Cost	Power Cons. For 1000 m^3/hr
Refrigeration	-20	Low	Low	2.9 kW
Desiccant regenerative (blower reactivated type)	-40	Medium	Medium	18.0 kW
Desiccant regenerative (heatless purge type)	-40	Low	High	20.7 kW
Desiccant regenerative (heat of compression type)	-40	High	Very low	0.8 kW

Air Receivers

The air receiver dampens pulsations entering the discharge from the compressor, serves as a reservoir for sudden or unusually heavy demands in excess of compressor capacity, prevents too frequent loading and unloading (short cycling) of the compressor and separates moisture and oil vapour, allowing the moisture carried over from the after coolers to precipitate.

The air receiver should be generously sized to give a large cooling surface and even out the pulsation in delivered air pressure from reciprocating compressor. As per IS 7938 – 1976, volume of the air receiver (in m^3) should be 1/10th to 1/6th of output in m^3/min. Providing an air receiver near the load end, where there is sudden high demand lasting for a short period, would avoid the need to provide extra compressor capacity.

Sizing of compressed air piping

The compressed air piping is sized based on the allowable velocity of compressed air in the pipeline, keeping a check on the pressure drop. In compressed air line if the pressure drop is high, the operating pressure at the generation end has to be increased to match with the requirement. This will result in increased power consumption of the compressor.

Velocities between 6 to 10 m/sec are usual in compressed air mains. This velocity range is sufficiently low to prevent excessive pressure drops on most systems and also will allow moisture to precipitate.

The volume of air changes if any of the following changes.

- Operating Pressure
- Temperature
- Both Pressure and Temperature

The law dealing with the expansion of air wherein pressure and temperature are changing together is given below.

$$P_1 * V_1 / T_1 = P_2 * V_2 / T_2$$

Where P_1, V_1 and T_1 are the original pressure, volume and temperature and P_2, V_2 and T_2 are the new pressure, volume and temperature respectively.

The following procedure can be adopted for estimating the size of a compressed air line.

Example

Quantity of compressed air flow	-	200 cfm
	-	5.66 m^3/min
Working pressure	-	7.0 kg/cm^2 (gauge)
Velocity limitation	-	6 m/sec
Apply gas laws $P_1 * V_1 / T_1$	=	$P_2 * V_2 / T_2$

Assuming the temperature remains constant.

$P_1 * V_1$	=	$P_2 * V_2$
V_2	=	$P_1 * V_1 / P_2$
	=	$\frac{5.66\ m^3/min \times 1.013\ kg/cm^2}{7.0\ kg/cm^2}$
		0.819 m^3/min
	=	0.01365 m^3/sec
Quantity of air flow	=	Area of pipe line x velocity of air flow
0.01365 m^3/sec	=	Area of pipe line x 6 m/sec
Area of pipe line	=	2.275 x 10^{-3} m^2
$\pi/4 \times D^2$	=	2.275 x 10^{-3} m^2
Diameter of pipeline	=	0.0538 m
	=	2.11"

Hence the diameter of pipeline can be chosen as 2"NB.

Loss of air pressure due to friction

The loss of pressure in piping is caused by resistance in pipe fittings and valves, which dissipates energy by producing turbulence. The piping system will be designed for a maximum allowable pressure drop of 5 percent from the compressor to the most distant point of use.

Piping layout

Where possible the piping system should be arranged as a closed loop or "ring main" to allow for more uniform air distribution to consumption points and to equalize pressure in the piping. Separate services requiring heavy air consumption and at long distances from the compressor unit should be supplied

by separate main airlines. Pipes are to be installed parallel with the lines of the building, with main and branch headers sloping down toward a dead end. Traps will be installed in airlines at all low points and dead ends to remove condensed moisture. Automatic moisture traps used for this purpose are effective only when the air has been cooled and the moisture has precipitated. Branch headers from compressed air mains will be taken off at the top to avoid picking up moisture.

Capacity Utilisation

In many installations, the use of air is intermittent. This means the compressor will be operated on low load or no load condition, which increases the specific power consumption per unit of air generated. Hence, for optimum energy consumption, a proper compressor capacity control should be selected. The nature of the control device depends on the function to be regulated. One of the objectives of a good compressed air management system would be to minimize unloading to the least as unloading consumes up to 30% of full load power.

One way of doing this is to use a smaller compressor.

Decentralized compressors, as against centralized compressors often serve this purpose better by having the option to switch off when air is not need in a particular section/equipment.

If a compressor is oversized and operates at unloading mode for long periods, an economical way will be to suitably change the pulley size of the motor or compressor and reduce the RPM to de-rate the compressor to a lower capacity.

With decreasing cost of variable speed drives, it has become a viable option to maintain constant pressure in the system and to avoid unloading operations by varying the speed of the compressor. However, caution should be taken for operations at very low speeds, since it will affect the lubricating system. This can be overcome by providing a separate lube oil system independent of the compressor.

3.6 Compressor Capacity Assessment

Due to ageing of the compressors and inherent inefficiencies in the internal components, the free air delivered may be less than the design value, despite good maintenance practices. Sometimes, other factors such as poor maintenance, fouled heat exchanger and effects of altitude also tend to reduce free air delivery. In order to meet the air demand, the inefficient compressor may have to run for more time, thus consuming more power than actually required.

The power wastage depends on the percentage deviation of FAD capacity. For example, a worn out compressor valve can reduce the compressor capacity by as much as 20 percent. A periodic assessment of the FAD capacity of each compressor has to be carried out to check its actual capacity. If the deviations are more than 10 %, corrective measures should be taken to rectify the same.

The ideal method of compressor capacity assessment is through a nozzle test wherein a calibrated nozzle is used as a load, to vent out the generated compressed air. Flow is assessed, based on the air temperature, stabilization pressure, orifice constant etc.

Simple method of Capacity Assessment in Shop floor

Isolate the compressor along with its individual receiver being taken for test from main compressed air system by tightly closing the isolation valve or blanking it, thus closing the receiver outlet.

Open water drain valve and drain out water fully and empty the receiver and the pipe line. Make sure that water trap line is tightly closed once again to start the test. Start the compressor and activate the stopwatch. Note the time taken to attain the normal operational pressure P_2 (in the receiver) from initial pressure P_1.

Calculate the capacity as per the formulae given below:

Actual Free air discharge

$$Q = \frac{P_2 - P_1}{P_0} \times \frac{V}{T} \text{ Nm}^3 / \text{Minute}$$

Where,

P_2	=	Final pressure after filling (kg/cm^2 a)
P_1	=	Initial pressure (kg/cm^{2}a) after bleeding
P_0	=	Atmospheric Pressure (kg/cm^2 a)
V	=	Storage volume in m^3 which includes receiver, after cooler, and delivery piping
T	=	Time take to build up pressure to P_2 in minutes

The above equation is relevant where the compressed air temperature is same as the ambient air temperature, i.e., perfect isothermal compression. In case the actual compressed air temperature at discharge, say $t_2{}^0$C is higher than ambient air temperature say $t_1{}^0$C (as is usual case), the FAD is to be corrected by a factor $(273 + t_1) / (273 + t_2)$.

EXAMPLE 1

An instrument air compressor capacity test gave the following results (assume the final compressed air temperature is same as the ambient temperature) – Comment?

Piston displacement	:	16.88 m^3/minute
Theoretical compressor capacity	:	14.75 m^3/minute @ 7 kg/cm^2
Compressor rated rpm 750	:	Motor rated rpm : 1445
Receiver Volume	:	7.79 m^3
Additional hold up volume, i.e., pipe / water cooler, etc., is	:	0.4974 m^3
Total volume	:	7.79+0.4974 = 8.287 m^3

Initial pressure P_1	:	0.5 kg/cm^2
Final pressure P_2	:	7.03 kg/cm^2
Atmospheric pressure P_0	:	1.026 kgf/cm^{2}A
Time taken to build up pressure	:	4.021 minutes
Compressor output m^3/minute	:	$\frac{(P_2 - P_1) \times \text{Total Volume}}{\text{Atm. Pressure} \times \text{Pump up time}}$

$$\frac{(7.03 - 0.5) \times 8.287}{1.026 \times 4.021} = 13.12 \text{ m}^3/\text{minute}$$

Capacity shortfall with respect to 14.75 m^3/minute rating is 1.63 m^3/minute i.e., 11.05 %, which indicates compressor performance needs to be investigated further.

3.7 Checklist for Energy Efficiency in Compressed Air System

- Ensure air intake to compressor is not warm and humid by locating compressors in well-ventilated area or by drawing cold air from outside. Every 4^{0}C rise in air inlet temperature will increase power consumption by 1 percent.
- Clean air-inlet filters regularly. Compressor efficiency will be reduced by 2 percent for every 250 mm WC pressure drop across the filter.
- Keep compressor valves in good condition by removing and inspecting once every six months. Worn-out valves can reduce compressor efficiency by as much as 50 percent.
- Install manometers across the filter and monitor the pressure drop as a guide to replacement of element.
- Minimize low-load compressor operation; if air demand is less than 50 percent of compressor capacity, consider change over to a smaller compressor or reduce compressor speed appropriately (by reducing motor pulley size) in case of belt driven compressors.
- Consider the use of regenerative air dryers, which uses the heat of compressed air to remove moisture.
- Fouled inter-coolers reduce compressor efficiency and cause more water condensation in air receivers and distribution lines resulting in increased corrosion. Periodic cleaning of inter-coolers must be ensured.
- Compressor free air delivery test (FAD) must be done periodically to check the present operating capacity against its design capacity and corrective steps must be taken if required.
- If more than one compressor is feeding to a common header, compressors must be operated in such a way that only one small compressor should handle the load variations whereas other compressors will operate at full load.
- The possibility of heat recovery from hot compressed air to generate hot air or water for process application must be economically analyzed in case of large compressors.
- Consideration should be given to two-stage or multistage compressor as it consumes less power for the same air output than a single stage compressor.

- If pressure requirements for processes are widely different (e.g. 3 bar to 7 bar), it is advisable to have two separate compressed air systems.
- Reduce compressor delivery pressure, wherever possible, to save energy.
- Provide extra air receivers at points of high cyclic-air demand which permits operation without extra compressor capacity.
- Retrofit with variable speed drives in big compressors, say over 100 kW, to eliminate the `unloaded' running condition altogether.
- Keep the minimum possible range between load and unload pressure settings.
- Automatic timer controlled drain traps wastes compressed air every time the valve opens. So frequency of drainage should be optimized.
- Check air compressor logs regularly for abnormal readings, especially motor current cooling water flow and temperature, inter-stage and discharge pressures and temperatures and compressor load-cycle.
- Compressed air leakage of 40- 50 percent is not uncommon. Carry out periodic leak tests to estimate the quantity of leakage.
- Install equipment interlocked solenoid cut-off valves in the air system so that air supply to a machine can be switched off when not in use.
- Present energy prices justify liberal designs of pipeline sizes to reduce pressure drops.
- Compressed air piping layout should be made preferably as a *ring main* to provide desired pressures for all users.
- A smaller dedicated compressor can be installed at load point, located far off from the central compressor house, instead of supplying air through lengthy pipelines.
- All pneumatic equipment should be properly lubricated, which will reduce friction, prevent wear of seals and other rubber parts thus preventing energy wastage due to excessive air consumption or leakage.
- Misuse of compressed air such as for body cleaning, agitation, general floor cleaning, and other similar applications must be discouraged in order to save compressed air and energy.
- Pneumatic equipment should not be operated above the recommended operating pressure as this not only wastes energy but can also lead to excessive wear of equipment's components which leads to further energy wastage.
- Pneumatic transport can be replaced by mechanical system as the former consumed about 8 times more energy. Highest possibility of energy savings is by reducing compressed air use.
- Pneumatic tools such as drill and grinders consume about 20 times more energy than motor driven tools. Hence they have to be used efficiently. Wherever possible, they should be replaced with electrically operated tools.
- Where possible welding is a good practice and should be preferred over threaded connections.
- On account of high pressure drop, ball or plug or gate valves are preferable over globe valves in compressed air lines.

Solved Example:

a) In an automobile industry a pump-up test was conducted to determine the free air delivery (FAD) of a reciprocating compressor and the following data were obtained:

Receiver capacity and additional holdup volume		
in piping and after-cooler	:	4100 litres
Initial pressure	:	1 kg/cm^2 (g)
Final pressure	:	8.5 kg/cm^2 (g)
Atmospheric Pressure	:	1.026 kg/cm^2 (a)
Ambient air temperature	:	32 ^oC
Final compressed air temperature	:	52 ^oC
Compressor pump up time	:	65 secs

Calculate the FAD of the compressor in cubic foot per minute.

b) Further a leakage test was carried out in the same compressed air system and with the same compressor as in problem a) above and following were the observations:

- Compressor was on load for 03 minutes
- Compressor was unloaded for 13 minutes
- Compressor was drawing 145 kW during load

Calculate the following:

i. % leakage in compressed air system

ii. Leakage quantity

iii. Specific power consumption

iv. Power lost due to leakage

Ans:

a) $$Q = \frac{P_2 - P_1}{P_0} \times \frac{V}{t} \times \left(\frac{273 + t_1}{273 + t_2} \right)$$

Time = 65 sec = 1.0833 secs

= (8.5 - 1)/1.026X 4.1 /1.0833) X (305/325)

= 25.96 m^3/min

= 25.96 x (3.28)^3

= 916 cfm

b)

i) % Leakage in the system

Load time (T) : 03 minutes

Unload time (t) : 13 minutes

% leakage in the system : $\frac{T}{(T+t)} x100$

: $\frac{3}{(3+13)} x100$ = 18.75 %

ii) Leakage quantity : 0.1875 x 916

: 171.75 cfm

iii) Operating capacity (FAD) : 916 cfm

Actual power consumption : 145 kW

Specific power consumption : 145/916

= 0.1583 kW/cfm

iv) Power lost due to leakage : leakage quantity x sp. power consumption

: 171.75 x 0.1583

: 27.19 kW

QUESTIONS

Objective Type Questions

1.	In a textile mill, two 150 cfm belt driven reciprocating compressors are seen to be working constantly with a loading time of 20 seconds and unloading time of 30 seconds. The best economic option for energy savings would be: a) switch off one compressor b) switch off one compressor and reduce motor pulley size of the other compressor appropriately c) adopt variable speed drive for one of the compressors d) none of the above
2.	Which of the following pipe fittings used in compressed air pipeline offers maximum resistance a) Gate Valve b) Tee 90 deg long bend c) Elbow d) Return bend
3.	Friction loss in a piping system carrying fluid is proportional to a) fluid flow b) (fluid flow)2 c) 1/fluid flow d) 1/(fluid flow)2
4.	The most energy intensive dryer among the following a) refrigeration b) desiccant (heat of compression) c) desiccant (heatless purge) d) desiccant (blower reactivated)
5.	The efficiency of compressed air system is around a) 90% b) 60% c) 50% d) 1 0 %
6.	The basic function of air dryer in a compressed air system is to a) remove remaining traces of moisture after the aftercooler b) store and smoothen pulsating air output c) reduce the temperature of the air before it enters the next state to increase efficiency d) prevent dust from entering compressor
7.	Select the correct statement for reciprocating air compressors: a) for every 4oC drop in the inlet air temperature, the increase in energy consumption is by 1%. b) for every 4oC rise in the inlet air temperature, the decrease in energy consumption is by 1% c) for every 4oC rise in the inlet air temperature, the increase in energy consumption is by 1% d) the energy consumption remains same irrespective of inlet air temperature
8.	Which of the following parameters is not required for evaluating volumetric efficiency of the compressor? a) FAD b) Cylinder bore diameter c) Stroke length d) Power input
9.	Which of the following will not occur if a reciprocating compressor is operated at a lower discharge pressure? a) lower power consumption b) less load on the piston rods and hence reduced maintenance costs c) lower leakage losses d) lower free air delivery than rated

10.	Which type of energy efficient dryer can be opted if a user in a plant requires compressed air at a dew point of -40°C ? a) heatless purge type dryer b) heat of compression dryer c) aftercooler d) refrigerant dryers
	Short Type Questions
S-1	Calculate the free air delivery (FAD) capacity of a compressor in m^3/hr for the following observed data: Receiver capacity : 0.3 m^3 Initial pressure (with empty receiver) : 0 kg/cm^2 (g) Final pressure : 7 kg/cm^2 (g) Initial air temperature : 35°C Final air temperature : 50 °C Additional holdup volume : 0.05 m^3 Compressor pump up time : 4.1 minutes Atmospheric pressure : 1.026 kg/cm^2 (a)
S-2	A reciprocating V belt driven compressor was found to operating during normal factory operation with the following parameters: Load pressure = 6 bar Unload pressure = 8 bar Load time = 3 minutes Unload time = 1.5 minutes Suggest possible energy saving opportunities on a short-term basis.
S-3	Briefly explain shop floor method of air compressor capacity assessment?
S-4	List down any 5 energy conservation opportunities in compressed air system
S-5	Explain how a tranvector nozzle works?
	Long Type Questions
L-1	a) In an automobile industry a pump-up test was conducted to determine the free air delivery (FAD) of a reciprocating compressor and the following data were obtained: Receiver capacity and additional holdup volume in piping and after-cooler : 4100 litres nitial pressure : 1 kg/cm^2 (g) Final pressure : 8.5 kg/cm^2 (g) Atmospheric Pressure : 1.026 kg/cm^2 (a) Ambient air temperature : 32 °C Final compressed air temperature : 52 °C Compressor pump up time : 65 secs Calculate the FAD of the compressor in cubic foot per minute.

	b) Further a leakage test was carried out in the same compressed air system and with the same compressor as in problem a) above and following were the observations: - Compressor was on load for 03 minutes - Compressor was unloaded for 13 minutes - Compressor was drawing 145 kW during load Calculate the following: i. % leakage in compressed air system ii. Leakage quantity iii. Specific power consumption iv. Power lost due to leakage
L-2	A free air delivery test was carried out before conducting a leakage test on a reciprocating air compressor in an engineering industry and following were the observations: Receiver capacity : 8.0 m^3 Initial pressure : 0.1 kg / cm^2g Final pressure : 7.0 kg / cm^2g Additional hold-up volume : 0.3 m^3 Atmospheric pressure : 1.026 kg / cm^2 abs. Compressor pump-up time : 3.5 minutes Further the following observations were made during the conduct of leakage test during the lunch time when no pneumatic equipment/ control valves were in operation: a) Compressor on load time is 24 seconds and unloading pressure is 7 kg/cm^2g b) Average power drawn by the compressor during loading is 92 kW c) Compressor unload time and loading pressure are 79 seconds and 6.6 kg/cm^2 g respectively. Find out the following: (i) Compressor output in m^3/hr(neglect temperature correction) (ii) Specific Power Consumption, kW/ m^3/hr (iii) % air leakage in the system (iv) leakage quantity in m^3/hr (v) power lost due to leakage

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Chapter 4

HVAC & Refrigeration System

Learning Objectives

In this chapter you will learn about

- Psychrometrics and air conditioning process
- Types and applications of HVAC and refrigeration system in industry
- Comparison of energy performance of various types of HVAC and refrigeration system
- Performance assessment of room air conditioners
- Performance assessment factors and energy efficiency
- Heat pumps and their applications
- Ice bank systems
- Humidification system
- Standards and labeling programme for air conditioners
- Energy saving opportunities

4. HVAC AND REFRIGERATION SYSTEM

Introduction, Psychrometrics and air conditioning processes, Types of refrigeration systems, Refrigerants and properties, Compressor types, Selection of refrigeration system, Performance assessment of room air conditioners (window and split), Factors affecting performance of refrigeration and air conditioning system, Energy savings opportunities. Cold storage, Heat pumps and their applications, Ventilation system, Ice bank system, Humidification system, Standards and labeling of air conditioners.

4.1 Introduction

The Heating, Ventilation and Air Conditioning (HVAC) and refrigeration system transfers the heat energy from or to the products, or building environment. Energy in form of electricity or heat is used to power mechanical equipment designed to transfer heat from a colder, low-energy level to a warmer, high-energy level.

Refrigeration deals with the transfer of heat from a low temperature level at the heat source to a high temperature level at the heat sink by using a low boiling refrigerant.

There are several heat transfer loops in refrigeration system as described below:

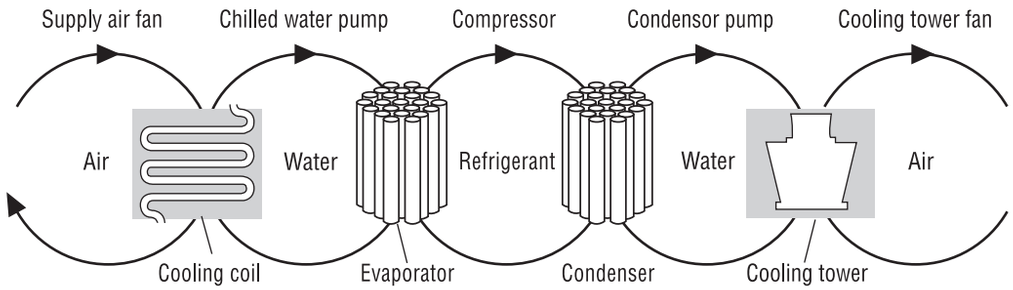


Figure 4.1 Heat Transfer Loops in Refrigeration System

In the Figure 4.1, thermal energy moves from left to right as it is extracted from the space and expelled into the outdoors through five loops of heat transfer:

- **Indoor air loop**. In the leftmost loop, indoor air is driven by the supply air fan through a cooling coil, where it transfers its heat to chilled water. The cool air then cools the building space.
- **Chilled water loop**. Driven by the chilled water pump, water returns from the cooling coil to the chiller's evaporator to be re-cooled.
- **Refrigerant loop**. Using a phase-change refrigerant, the chiller's compressor pumps heat from the chilled water to the condenser water.
- **Condenser water loop**. Water absorbs heat from the chiller's condenser, and the condenser water pump sends it to the cooling tower.
- **Cooling tower loop**. The cooling tower's fan drives air across an open flow of the hot condenser water, transferring the heat to the outdoors.

4.2 Psychrometrics and Air-Conditioning Processes

Psychrometrics is the science of moist air properties and processes, which is used to illustrate and analyze air-conditioning cycles. It translates the knowledge of heating or cooling loads (which are in kW or tons) into volume flow rates (in m^3/s or cfm) for the air to be circulated into the duct system.

Water vapor is lighter than dry air. The amount of water vapor that the air can carry increases with its temperature. Any amount of moisture that is present beyond what the air can carry at the prevailing temperature can only exist in the liquid phase as suspended liquid droplets (if the air temperature is above the freezing point of water), or in the solid state as suspended ice crystals (if the temperature is below the freezing point).

The most commonly used psychrometric quantities include the dry and wet bulb temperatures, dew point, specific humidity, relative humidity.

Psychrometric Chart:

Psychrometric chart (Figure 4.2) is a chart indicating the psychrometric properties of air such as dry-bulb temperature, wet-bulb temperature, specific humidity, enthalpy of air in kJ/kg dry air, specific volume of air in m^3/kg and relative humidity ø in %. It helps in quantifying and understanding air conditioning process.

Example: Assume that the outside air temperature is 32°C with a relative humidity of ϕ= 60%. Use the psychrometric chart to determine the air properties? See Figure 4.2.

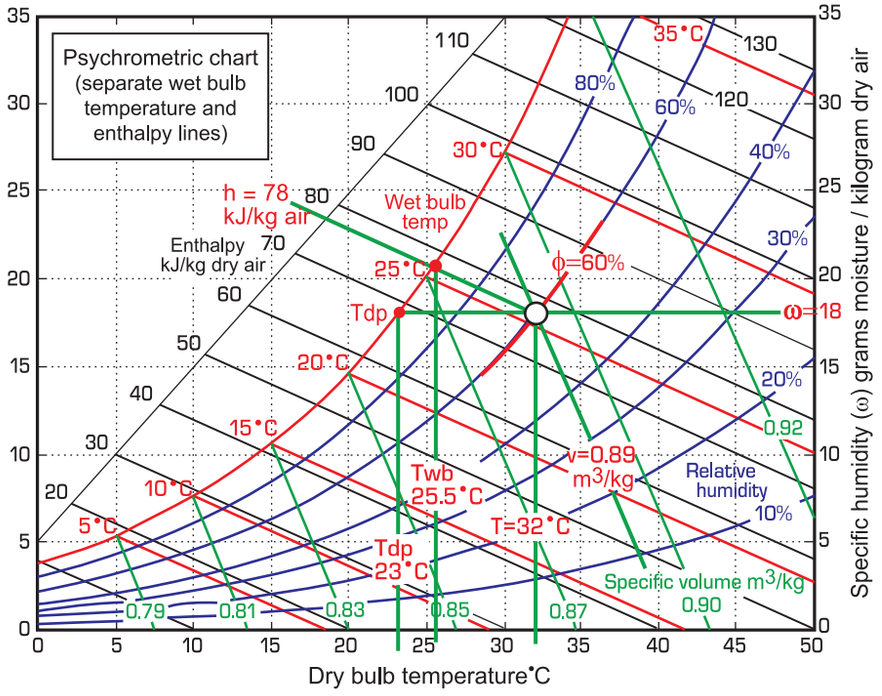


Figure 4.2 Properties of Air at 32°C Dry Bulb Temperature and RH of 60%

Solution

Air properties of air at 32°C dry bulb temperature and RH of 60 %

Specific humidity ω	=	18 gm-moisture/kg-air
Enthalpy h	=	78 kJ/kg-air
Wet-bulb temperature T_{wb}	=	25.5°C
Dew-point temperature T_{dp}	=	23°C
The specific volume of the dry air v	=	0.89m^3/kg

Comfort Zone

One of the major applications of the Psychrometric Chart is in air conditioning, and we find that most humans feel comfortable when the temperature is between 22°C and 27°C, and the relative humidity ϕ between 40% and 60%. This defines the "comfort zone" which is portrayed on the Psychrometric Chart as shown in Figure 4.3. Thus with the aid of the chart we either heat or cool, add moisture or dehumidify as required in order to bring the air into the comfort zone.

Example 2: Outside air at 35°C and 60% relative humidity is to be conditioned by cooling and re-heating so as to bring the air to the "comfort zone" with the exit temperature of 24°C and 53% RH. Using the Psychrometric Chart neatly plot the required air conditioning process and estimate (a) the amount of moisture removed, (b) the heat removed, and (c) the amount of heat added. See Figure 4.3.

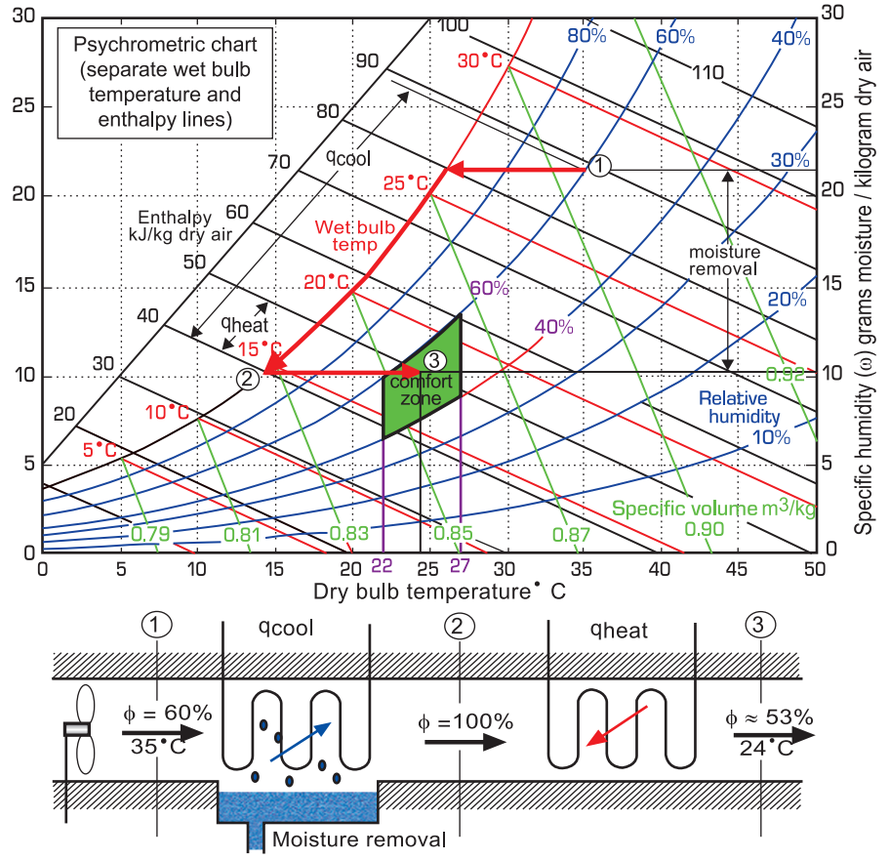


Figure 4.3 Properties of Air

Ans: Using the Figure 4.3

(a) The amount of moisture removed = 11.5g-H_2O /kg-dry-air

(b) The heat removed = (1)-(2), q_{cool} = 48 kJ/kg-dry-air

(c) The amount of heat added = (2)-(3), q_{heat} = 10 kJ/kg-dry-air

Air-Conditioning Systems

Depending on applications, there are several options / combinations, which are available for use as given below:

- Air Conditioning (for comfort / machine)
- Split air conditioners
- Fan coil units in a larger system
- Air handling units in a larger system

Refrigeration Systems (for processes)

- Small capacity modular units of direct expansion type similar to domestic refrigerators, small capacity refrigeration units.
- Centralized chilled water plants with chilled water as a secondary coolant for temperature range over 5 ^{0}C typically. They can also be used for ice bank formation.
- Brine plants, which use brines as lower temperature, secondary coolant, for typically sub zero temperature applications, which come as modular unit capacities as well as large centralized plant capacities.
- The plant capacities upto 50 TR are usually considered as small capacity, 50 – 250 TR as medium capacity and over 250 TR as large capacity units.

A large industry may have a bank of such units, often with common chilled water pumps, condenser water pumps, cooling towers, as an off site utility.

The same industry may also have two or three levels of refrigeration & air conditioning such as:

- Comfort air conditioning (20 – 25 ^{0}C)
- Chilled water system (8 – 10 ^{0}C)
- Brine system (sub-zero applications)

Two principle types of refrigeration plants found in industrial use are: Vapour Compression Refrigeration (VCR) and Vapour Absorption Refrigeration (VAR). VCR uses mechanical energy as the driving force for refrigeration, while VAR uses thermal energy as the driving force for refrigeration.

4.3 Types of Refrigeration System

Vapour Compression Refrigeration

Heat flows naturally from a hot to a colder body. In refrigeration system the opposite must occur i.e. heat flows from a cold to a hotter body. This is achieved by using a substance called a refrigerant,

which absorbs heat and hence boils or evaporates at a low pressure to form a gas. This gas is then compressed to a higher pressure, such that it transfers the heat it has gained to ambient air or water and turns back (condenses) into a liquid. In this way heat is absorbed, or removed, from a low temperature source and transferred to a higher temperature source.

The refrigeration cycle can be broken down into the following stages (see Figure 4.4):

1 - 2 Low pressure liquid refrigerant in the evaporator absorbs heat from its surroundings, usually air, water or some other process liquid. During this process it changes its state from a liquid to a gas, and at the evaporator exit is slightly superheated.

2 - 3 The superheated vapour enters the compressor where its pressure is raised. There will also be a big increase in temperature, because a proportion of the energy input into the compression process is transferred to the refrigerant.

3 - 4 The high pressure superheated gas passes from the compressor into the condenser. The initial part of the cooling process (3 - 3a) desuperheats the gas before it is then turned back into liquid (3a - 3b). The cooling for this process is usually achieved by using air or water. A further reduction in temperature happens in the pipe work and liquid receiver (3b - 4), so that the refrigerant liquid is sub-cooled as it enters the expansion device.

4 - 1 The high-pressure sub-cooled liquid passes through the expansion device, which both reduces its pressure and controls the flow into the evaporator.

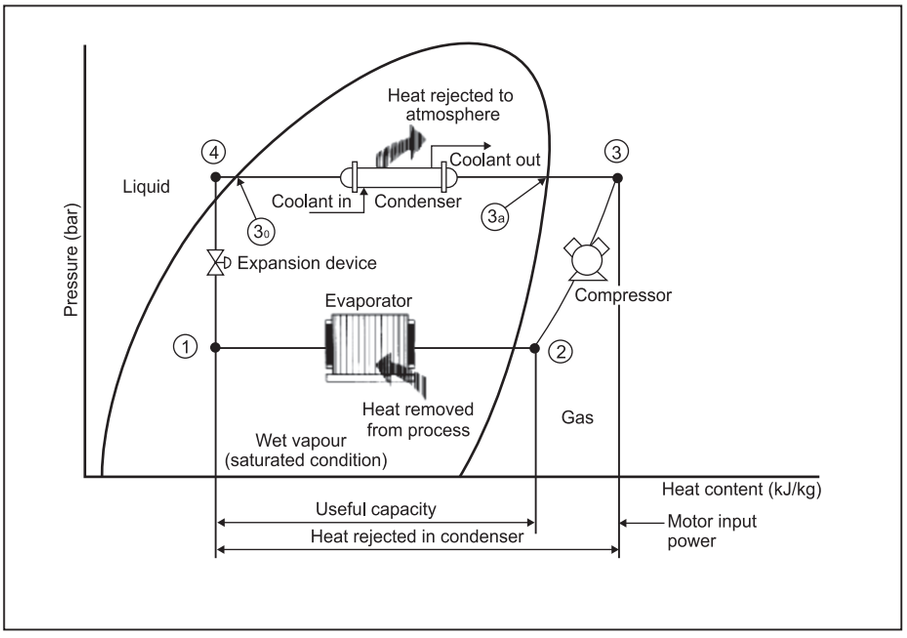


Figure 4.4 Schematic of a Basic Vapor Compression Refrigeration System

It can be seen that the condenser has to be capable of rejecting the combined heat inputs of the evaporator and the compressor; i.e. (1 - 2) + (2 - 3) has to be the same as (3 - 4). There is no heat loss or gain through the expansion device.

Alternative Refrigerants for Vapour Compression Systems

The use of CFCs is phased out due to their damaging impact on the protective tropospheric ozone layer around the earth. The Montreal Protocol of 1987 and the subsequent Copenhagen agreement of 1992 mandate a reduction in the production of ozone depleting Chlorinated Fluorocarbon (CFC) refrigerants in a phased manner, with an eventual stop to all production by the year 1996. As part of the accelerated phase-out of CFCs, India has completely phased out CFCs by 1st August. 2008.

In response, the refrigeration industry has developed two alternative refrigerants; one based on Hydrochloro Fluorocarbon (HCFC), and another based on Hydro Fluorocarbon (HFC). The HCFCs have a 2 to 10% ozone depleting potential as compared to CFCs and also, they have an atmospheric lifetime between 2 to 25 years as compared to 100 or more years for CFCs (Brandt, 1992). However, even HCFCs are mandated to be phased out, and only the chlorine free (zero ozone depletion) HFCs would be acceptable.

The 19th MOP (Meeting of Parties) took a decision to accelerate the phase-out of HCFC production and consumption for developed and developing countries. The new phase-out schedule for Article 5 parties as per the decision taken at the 19th MOP is as follows:

Base-level for production & consumption: the average of 2009 and 2010

Freeze= 2013 at the base-level

10% reduction in 2015

35% reduction in 2020

67.5% reduction in 2025

100% reduction in 2030 with a service tail of 2.5% annual average during the period 2030-2040(Source: Ministry of Environment and Forest, Ozone Cell)

Until now, only one HFC based refrigerant, HFC 134a, has been developed. HCFCs are comparatively simpler to produce and the three refrigerants 22, 123, and 124 have been developed. The use of HFCs and HCFCs results in slightly lower efficiencies as compared to CFCs, but this may change with increasing efforts being made to replace CFCs.

Absorption Refrigeration

The absorption chiller is a machine, which produces chilled water by using heat such as steam, hot water, gas, oil etc. Chilled water is produced by the principle that liquid (refrigerant), which evaporates at low temperature, absorbs heat from surrounding when it evaporates. Pure water is used as refrigerant and lithium bromide solution is used as absorbent

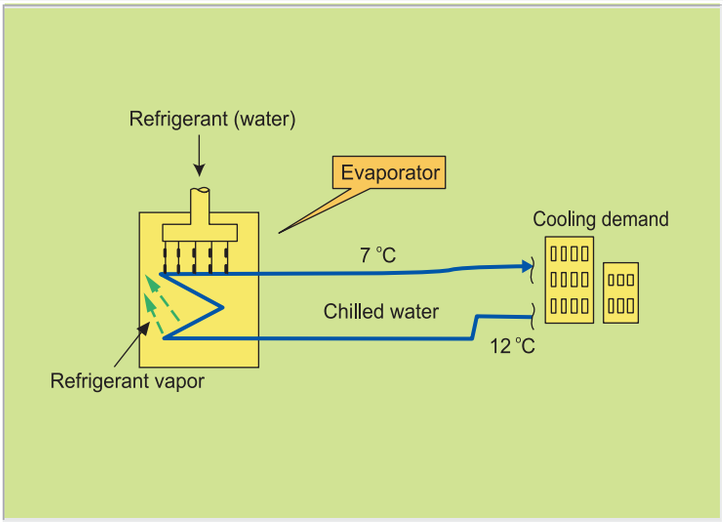
Heat for the vapour absorption refrigeration system can be provided by waste heat extracted from process, diesel generator sets etc. Absorption systems require electricity to run pumps only. Depending on the temperature required and the power cost, it may even be economical to generate heat / steam to operate the absorption system.

Description of the absorption refrigeration concept is given below:

The refrigerant (water) evaporates at around 4 ^{0}C under the high vacuum condition of 754 mmHg in the evaporator. When the refrigerant (water) evaporates, the latent heat of vaporization takes the heat from incoming chilled water.

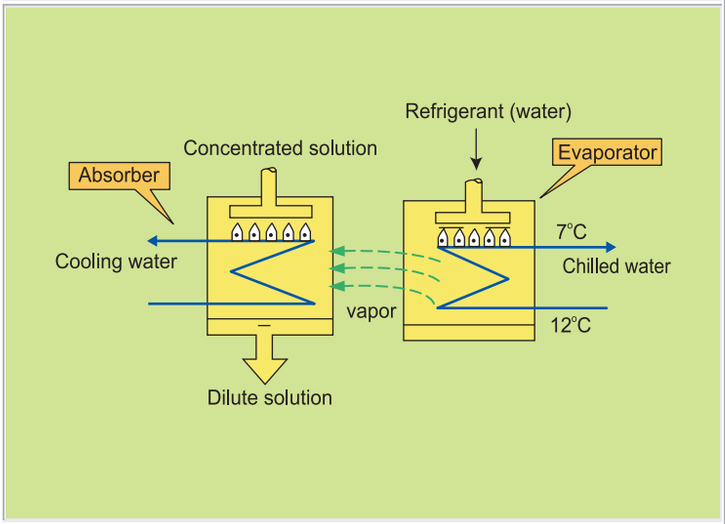
This latent heat of vaporization can cool the chilled water which runs into the heat exchanger tubes in the evaporator by transfer of heat to the refrigerant (water).

EVAPORATOR



In order to keep evaporating, the refrigerant vapor must be discharged from the evaporator and refrigerant (water) must be supplied. The refrigerant vapor is absorbed into lithium bromide solution which is convenient to absorb the refrigerant vapor in the absorber. The heat generated in the absorption process is led out of system by cooling water continually. The absorption also maintains the vacuum inside the evaporator.

ABSORBER

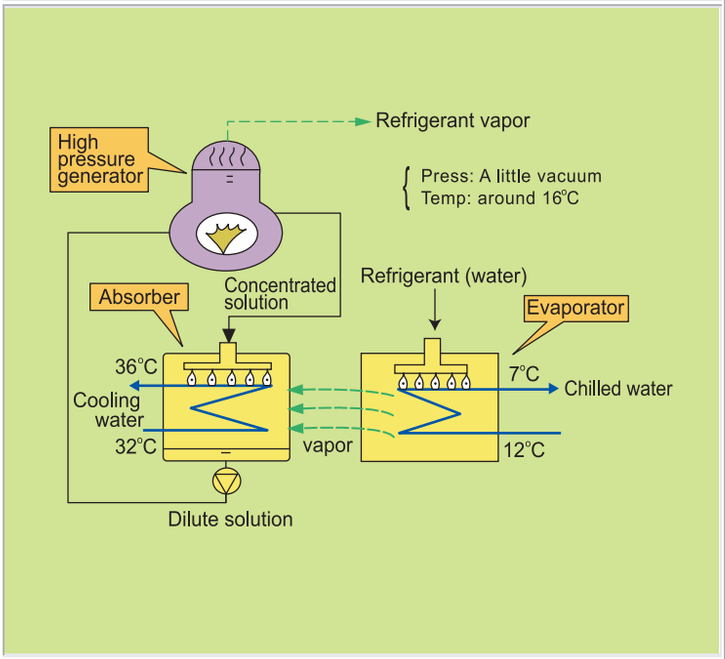


As lithium bromide solution is diluted, the effect to absorb the refrigerant vapor reduces. In order to keep absorption process, the diluted lithium bromide solution must be made concentrated lithium bromide.

Absorption chiller is provided with the solution concentrating system by the heating media such as steam, hot water, gas, oil, which performs such function is called generator.

The concentrated solution flows into the absorber and absorbs the refrigerant vapor again.

HIGH PRESSURE GENERATOR



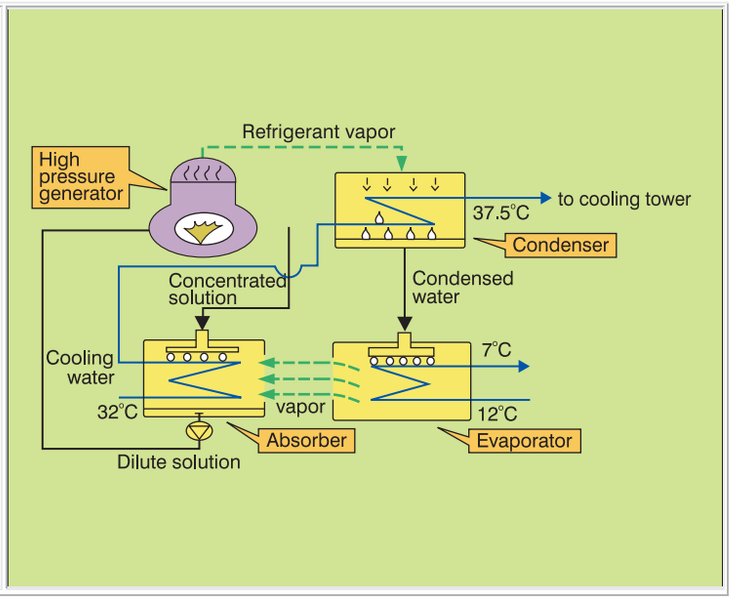
In order to carryout above works continually and to make complete cycle, the following two functions are required.

1. To concentrate and liquefy the evaporated refrigerant vapor, which is generated in the high pressure generator.

2. To supply the condensed water to the evaporator as refrigerant (water).

For these function, condenser is installed.

CONDENSER



A typical schematic of the absorption refrigeration system is given in the Figure 4.5.

Li-Br-water absorption refrigeration systems have a Coefficient of Performance (COP) in the range of 0.65 - 0.70 and can provide chilled water at 6.7 °C with a cooling water temperature of 30 °C. Systems capable of providing chilled water at 3 °C are also available. Ammonia based systems operate at above atmospheric pressures and are capable of low temperature operation (below 0°C). Absorption machines of capacities in the range of 10-1500 tons are available. Although the initial cost of absorption system is higher than compression system, operational cost is much lower-if waste heat is used.

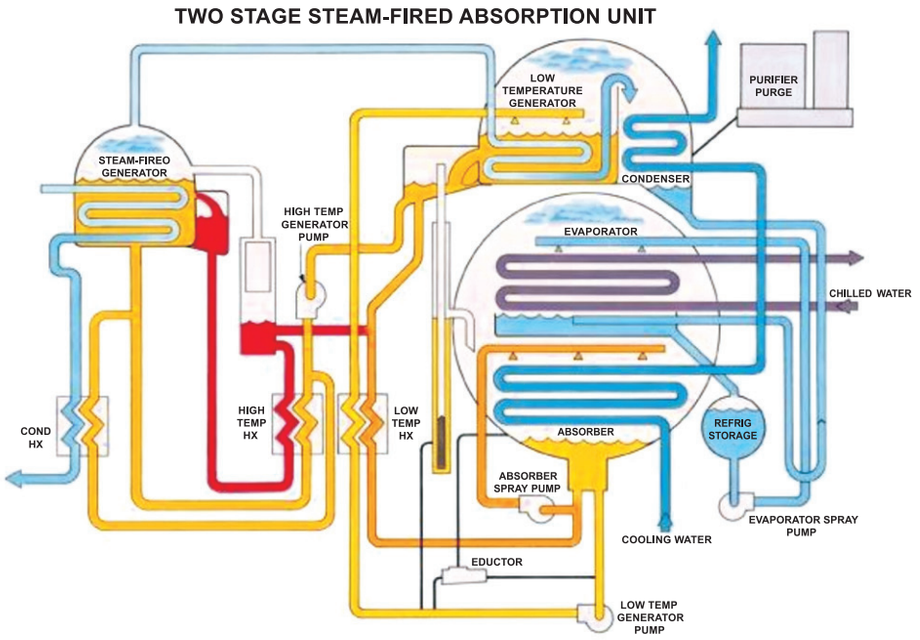


Figure 4.5 Schematic of Absorption Refrigeration System

Evaporative Cooling

There are occasions where air conditioning, which stipulates control of humidity up to 50 % for human comfort or for process, can be replaced by a much cheaper and less energy intensive evaporative cooling.

The concept is very simple and is the same as that used in a cooling tower. Air is brought in close contact with water to cool it to a temperature close to the wet bulb temperature. The cool air can be used for comfort or process cooling. The disadvantage is that the air is rich in moisture. Nevertheless, it is an extremely efficient means of cooling at very low cost. Large commercial systems employ cellulose filled pads over which water is sprayed. The temperature can be controlled by controlling the airflow and the water circulation rate. The possibility of evaporative cooling is especially attractive for comfort cooling in dry regions. This principle is practiced in textile industries for certain processes.

4.4 Common Refrigerants and Properties

A variety of refrigerants are used in vapor compression systems. The choice of fluid is determined largely by the cooling temperature required. Commonly used refrigerants are in the family of chlorinated fluorocarbons (CFCs, also called Freons): R-11, R-12, R-21, R-22 and R-502. The properties of these refrigerants are summarized in Table 4.1 and the performance of these refrigerants is given in Table 4.2.

Table 4.1 Properties of Commonly used Refrigerants

Refrigerant	Boiling Point ** (°C)	Freezing Point (°C)	Vapor Pressure * (kPa)	Vapor Volume * (m^3 / kg)	Enthalpy *	
					Liquid (kJ / kg)	Vapor (kJ / kg)
R - 11	-23.82	-111.0	25.73	0.61170	191.40	385.43
R - 12	-29.79	-158.0	219.28	0.07702	190.72	347.96
R - 22	-40.76	-160.0	354.74	0.06513	188.55	400.83
R - 502	-45.40	---	414.30	0.04234	188.87	342.31
R - 7 (Ammonia)	-33.30	-77.7	289.93	0.41949	808.71	487.76

* *At -10 °C* ** *At Standard Atmospheric Pressure (101.325 kPa)*

Table 4.2 Performance of Commonly used Refrigerants*

Refrigerant	Evaporating Press (kPa)	Condensing Press (kPa)	Pressure Ratio	Vapor Enthalpy (kJ / kg)	COP^{**}_{carnot}
R - 11	20.4	125.5	6.15	155.4	5.03
R - 12	182.7	744.6	4.08	116.3	4.70
R - 22	295.8	1192.1	4.03	162.8	4.66
R - 502	349.6	1308.6	3.74	106.2	4.37
R - 717	236.5	1166.5	4.93	103.4	4.78

* *At -15 °C Evaporator Temperature, and 30 °C Condenser Temperature*

** COP_{carnot} = *Coefficient of Performance* = $Temp._{Evap.} / (Temp._{Cond.} - Temp_{Evap.})$

The choice of refrigerant and the required cooling temperature and load determine the choice of compressor, as well as the design of the condenser, evaporator, and other auxiliaries. Additional factors such as ease of maintenance, physical space requirements and availability of utilities for auxiliaries (water, power, etc.) also influence component selection.

4.5 Compressor Types and Application

For industrial use, open type systems (compressor and motor as separate units) are normally used, though hermetic systems (motor and compressor in a sealed unit) also find service in some low capacity applications. Hermetic systems are used in refrigerators, air conditioners, and other low capacity applications. Industrial applications largely employ reciprocating, centrifugal and, more recently, screw compressors, and scroll compressors. Water-cooled systems are more efficient than air-cooled alternatives because the temperatures produced by refrigerant condensation are lower with water than with air.

Centrifugal Compressors

Centrifugal compressors are the most efficient type (see Figure 4.6) when they are operating near full load. Their efficiency advantage is greatest in large sizes, and they offer considerable economy of scale, so they dominate the market for large chillers. They are able to use a wide range of refrigerants efficiently, so they will probably continue to be the dominant type in large sizes.

Figure 4.6 Centrifugal Compressor

Centrifugal compressors have a single major moving part - an impeller that compresses the refrigerant gas by centrifugal force. The gas is given kinetic energy as it flows through the impeller. This kinetic energy is not useful in itself, so it must be converted to pressure energy. This is done by allowing the gas to slow down smoothly in a stationary diffuser surrounding the impeller.

To minimize efficiency loss at reduced loads, centrifugal compressors typically throttle output with inlet guide vanes located at the inlet to the impeller(s). This method is efficient down to about 50% load, but the efficiency of this method decreases rapidly below 50% load.

Older centrifugal machines are not able to reduce load much below 50%. This is because of "surge" in the impeller. As the flow through the impeller is choked off, the gas does not acquire enough energy to overcome the discharge pressure. Flow drops abruptly at this point, and an oscillation begins as the gas *flutters* back and forth in the impeller. Efficiency drops abruptly, and the resulting vibration can damage the machine. Many older centrifugal machines deal with low loads by creating a false load on the system, such as by using hot gas bypass. This wastes the portion of the cooling output that is not required.

Another approach is to use variable-speed drives in combination with inlet guide vanes. This may allow the compressor to throttle down to about 20% of full load, or less, without false loading. Changing the impeller speed causes a departure from optimum performance, so efficiency still declines badly at low loads. A compressor that uses a variable-speed drive reduces its output in the range between full load and approximately half load by slowing the impeller speed. At lower loads, the impeller cannot

be slowed further, because the discharge pressure would become too low to condense the refrigerant. Below the minimum load provided by the variable-speed drive, inlet guide vanes are used to provide further capacity reduction.

Reciprocating Compressors

The maximum efficiency of reciprocating compressors (see Figure 4.7) is lower than that of centrifugal and screw compressors. Efficiency is reduced by clearance volume (the compressed gas volume that is left at the top of the piston stroke), throttling losses at the intake and discharge valves, abrupt changes in gas flow, and friction. Lower efficiency also results from the smaller sizes of reciprocating units, because motor losses and friction account for a larger fraction of energy input in smaller systems.

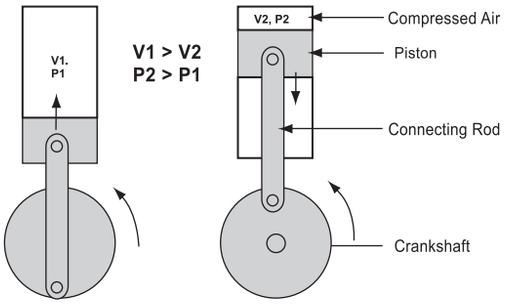


Figure 4.7 Reciprocating Compressor

Reciprocating compressors suffer less efficiency loss at partial loads than other types, and they may actually have a higher absolute efficiency at low loads than the other types. Smaller reciprocating compressors control output by turning on and off. This eliminates all part-load losses, except for a short period of inefficient operation when the machine starts.

Larger multi-cylinder reciprocating compressors commonly reduce output by disabling ("unloading") individual cylinders. When the load falls to the point that even one cylinder provides too much capacity, the machine turns off. Several methods of cylinder unloading are used, and they differ in efficiency. The most common is holding open the intake valves of the unloaded cylinders. This eliminates most of the work of compression, but a small amount of power is still wasted in pumping refrigerant gas to-and-fro through the unloaded cylinders. Another method is blocking gas flow to the unloaded cylinders, which is called "suction cutoff."

Variable-speed drives can be used with reciprocating compressors, eliminating the complications of cylinder unloading. This method is gaining popularity with the drastic reduction in costs of variable speed drives.

Screw Compressors

Screw compressors, sometimes called "helical rotary" compressors, compress refrigerant by trapping it in the "threads" of a rotating screw-shaped rotor (see Figure 4.8). Screw compressors have increasingly taken over from reciprocating compressors of medium sizes and large sizes, and they have even entered the size domain of centrifugal machines. Screw compressors are applicable to refrigerants that have higher condensing pressures, such as HCFC-22 and ammonia. They are especially compact. A variety of methods are used to control the output of screw compressors. There are major efficiency differences among the different methods. The most common is a slide valve that forms a portion of the housing that surrounds the screws.

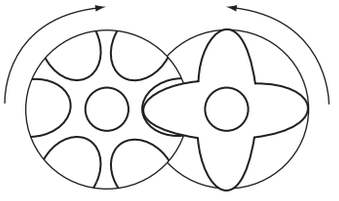


Figure 4.8 Screw Compressor

Using a variable-speed drive is another method of capacity control. It is limited to oil-injected compressors, because slowing the speed of a dry compressor would allow excessive internal leakage. There are other methods of reducing capacity, such as suction throttling that are inherently less efficient than the previous two.

Scroll Compressors

The scroll compressor is an old invention that has finally come to the market. The gas is compressed between two scroll-shaped vanes. One of the vanes is fixed, and the other moves within it. The moving vane does not rotate, but its center revolves with respect to the center of the fixed vane, as shown in Figure 4.9. This motion squeezes the refrigerant gas along a spiral path, from the outside of the vanes toward the center, where the discharge port is located. The compressor has only two moving parts, the moving vane and a shaft with an off-center crank to drive the moving vane. Scroll compressors have only recently become practical, because close machining tolerances are needed to prevent leakage between the vanes, and between the vanes and the casing.

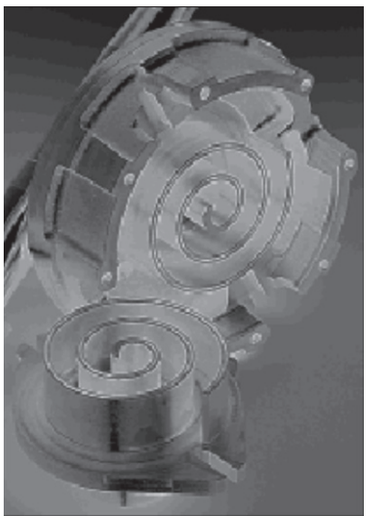


Figure 4.9 Scroll Compressor

The features of various refrigeration compressors and application criteria are given in the Table 4.3.

4.6 Selection of a Suitable Refrigeration System

A clear understanding of the cooling load to be met is the first and most important part of designing / selecting the components of a refrigeration system. Important factors to be considered in quantifying the load are the actual cooling need, heat (cool) leaks, and internal heat sources (from all heat generating equipment). Consideration should also be given to process changes and / or changes in ambient conditions that might affect the load in the future. Reducing the load, e.g. through better insulation, maintaining as high a cooling temperature as practical, etc. is the first step toward minimizing electrical power required to meet refrigeration needs. With a quantitative understanding of the required temperatures and the maximum, minimum, and average expected cooling demands, selection of appropriate refrigeration system (single-stage / multi-stage, economized compression, compound / cascade operation, direct cooling / secondary coolants) and equipment (type of refrigerant, compressor, evaporator, condenser, etc.) can be undertaken.

Table 4.3 Comparison of Different Types of Refrigeration Plants

(Source : ASHRAE & Vendor Information)

S. No.	Parameters	Vapour Compression Chillers			Vapour Absorption Chiller				
					LiBr				Ammonia
		Reciprocating	Centrifugal	Screw	Single Effect	Double Effect	Half Effect	Triple Effect	Single Stage
1	Refrigeration Temp. Range (Brine / Water)	+7 to -30^{o}C	+7 to -0^{o}C	+7 to -25^{o}C	Above 6^{0}C				Upto –33^{0}C
2	Energy Input	Electricity	Electricity	Electricity	Heat (Steam / Hot Water / Hot Oil/ Direct Fired)	Heat (Steam / Hot Water / Hot Oil/ Direct Fired)	Heat (Hot Water)	Heat (Steam /Hot Oil / Direct Fired)	Heat (Steam/ Hot Water / Hot Oil)
3	Heat Input Temp. Range - Maximum -Minimum	-	-	-	Minimum 85^{o}C	Minimum 130^{o}C	Minimum 55^{o}C	Minimum 190^{o}C	Minimum 85^{o}C
4	Typical Energy to TR Ratio								
	Air Conditioning Temp. Range	0.7-0.9 kW/TR	0.63kW/TR	0.65 kW/TR	5000 kcal/TR	2575 kcal/TR	7500 kcal/TR	2000 kcal/TR	4615 kcal/TR
	Subzero Temp. Range	1.25 to 2.5 kW/TR	-----	1.25 to 2.5 kW/TR	-----	-----	-----	-----	6666 kcal/hr
5	Refrigerant	R11,R123,R134a Ammonia	R22, R12	R22, R134a Ammonia	Pure Water	Pure Water	Pure Water	Pure Water	Pure Ammonia
6	Absorbent	------	------	------	Water-LiBr solution	Water-LiBr solution	Water-LiBr solution	Water-LiBr solution	Ammonia-LiBr solution
7	Typical single unit capacity range								
	Air Condition temp. range	1-150 TR	300 TR & above	50-200 TR	30 TR & above	30 TR & above	30 TR & above	50 TR & above	30 TR & above
	Subzero temp. range	10-50 TR	------	50-200 TR	-----	-----	-----	-----	30 TR & above
8	Typical COP at Part Load up to 50%	Reduces at part load	Reduces at part load	Improves by 15-20%	Marginal Improvement at Part Load				No variation
9	Typical Internal Pressure Levels - Low -High Typical Internal Temp. Levels	0.15-0.40 bar a 1.20-1.50 bar a -25 to 50^{o}C	2.5-3.5 bar a 11-12 bar -5 to 50^{o}C	2-5.5 bar 18-20 bar -25 to 50^{o}C	5-6 mm Hg (abs) 60-70 mm Hg (abs) +4 to 75^{o}C	5-6 mm Hg (abs) 370-390 mm Hg (abs) +4 to +130^{o}C	5-6 mm Hg (abs) 60-70 mm Hg (abs) +4 to 130^{o}C	5-6 mm Hg (abs) 2 kg/cm^{2} (a) +4 to 160^{o}C	1.2kg/cm^{2}(a) 18 kg/cm2(a) -25 to +150^{o}C

S. No.	Parameters	Vapour Compression Chillers			Vapour Absorption Chiller				
					LiBr				Ammonia
		Reciprocating	Centrifugal	Screw	Single Effect	Double Effect	Half Effect	Triple Effect	Single Stage
10	-Typical Cooling tower capacity range per 100 TR of chillers - Air-conditioning Temperature Range - Subzero temp. range	130 190	120 -----	120 160	260 ---	200 ---	370 ---	170 ---	290 290
11	Typical Make-up water quantity range in Ltrs/Hr. -Air Conditioning temperature range -Subzero temp. range	672 983	620 ---	620 830	1345 ---	1035 ---	1914 ---	880 ---	1500 1500
12	Material of construction -Generator	---	---	---	Cu-Ni or Stainless Steel				Carbon Steel
	-Absorber	---	---	---	Cu-Ni				Carbon Steel
	-Evaporator	Copper / Carbon steel	copper / Carbon steel	Copper/ Carbon steel	Cu-Ni				Carbon Steel
	-Condenser	Copper / Carbon steel	Copper / Carbon steel	Copper / Carbon steel	Cu-Ni				Carbon Steel
	-Solution Heat Exchange	---	---	---	Carbon Steel				Carbon Steel
	-Solution Pump	---	---	---	Cast Iron Hermatically Sealed (Canned motor type)				Cast Iron with Meh.Seal
	-Refrigerant pump	---	---	---	Cast Iron Hermatically Sealed (Canned motor type)				Not needed
13	Expected Life	25-30 years			15-20 years				50 years
14	Normally Expected Repairs / Maintenance	Periodic Compressor Overhaul Tube Replacement after 1-12 years			Tube Replacement due to Corrosion				Practically no repairs
15	Factory Assembled packaged Or Site Assembled	Factory Assembled							Factory Assembled upto 230 TR in A/C & subzero range
16	Beneficial Energy Sources	Low cost Electricity	Low cost Electricity	Low cost Electricity	a) Waste Heat b) Low cost steam / Low cost fuels				

S. No.	Parameters	Vapour Compression Chillers			Vapour Absorption Chiller				
					LiBr				Ammonia
		Reciprocating	Centrifugal	Screw	Single Effect	Double Effect	Half Effect	Triple Effect	Single Stage
17	Critical Parameters	-Electricity supply	-Lubrication System -Compressor Operation & Maintenance -Electrical Power Panel Maintenance	----	a) Vacuum in Chiller b) Purge System for Vacuum c) Corrosion Inhibitors in Absorbent d) Surfactants in Absorbent e) Cooling Water Treatment f) Cooling Water Temperature g) Heat Source Temperature				Sudden Power failure for 45-60 min. or more can disturb the distillation column for continuous operation. Needs D.G. set if there is frequent power failure for periods longer than 30 min.

4.7 Performance Assessment of Refrigeration Plants

The cooling effect produced is quantified as tons of refrigeration.(TR).

1 TR of refrigeration = 3024 kcal/hr heat rejected.

The refrigeration TR is assessed as TR = Q x×C_p x× (T_i – T_o) / 3024

Where,

Q is mass flow rate of coolant in kg/hr

C_p is coolant specific heat in kcal /kg deg C

T_i is inlet, temperature of coolant to evaporator (chiller) in ^{0}C

T_o is outlet temperature of coolant from evaporator (chiller) in ^{0}C

The above TR is also called as chiller tonnage.

The specific power consumption kW/TR is a useful indicator of the performance of refrigeration system. By measuring refrigeration duty performed in TR and the kiloWatt inputs, kW/TR is used as a reference energy performance indicator.

In a centralized chilled water system, apart from the compressor unit, power is also consumed by the chilled water (secondary) coolant pump as well condenser water (for heat rejection to cooling tower) pump and cooling tower fan in the cooling tower. Effectively, the overall energy consumption would be towards:

- Compressor kW
- Chilled water pump kW
- Condenser water pump kW
- Cooling tower fan kW, for induced / forced draft towers

The specific power consumption for certain TR output would therefore have to include:

- Compressor kW/TR
- Chilled water pump kW/TR
- Condenser water pump kW/TR
- Cooling tower fan kW/TR

The overall kW/TR is the sum of the above.

The theoretical Coefficient of Performance (Carnot), COP_{Carnot} - a standard measure of refrigeration efficiency of an ideal refrigeration system- depends on two key system temperatures, namely, evaporator temperature T_e and condenser temperature T_c with COP being given as:

$$COP_{Carnot} = T_e / (T_c - T_e)$$

This expression also indicates that higher COP_{Carnot} is achieved with higher evaporator temperature and lower condenser temperature.

But COP_{Carnot} is only a ratio of temperatures, and hence does not take into account the type of compressor. Hence the COP normally used in the industry is given by

$$COP = \frac{\text{Cooling effect (kW)}}{\text{Power input to compressor (kW)}}$$

Where the cooling effect is the difference in enthalpy across the evaporator and expressed as kW. The effect of evaporating and condensing temperatures are given in the Figure 4.10 and Figure 4.11 below:

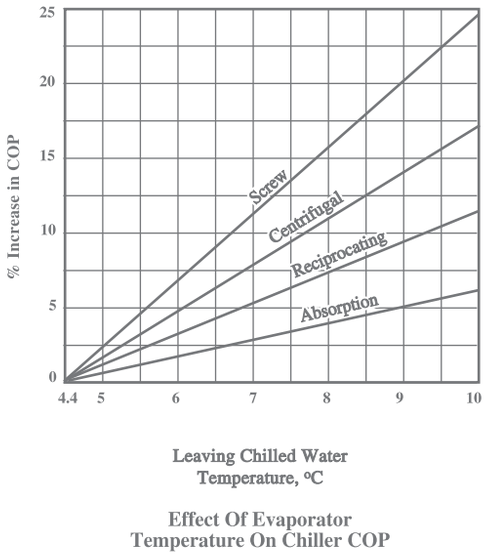


Figure 4.10 Effect of Evaporator Temperature on Chiller COP

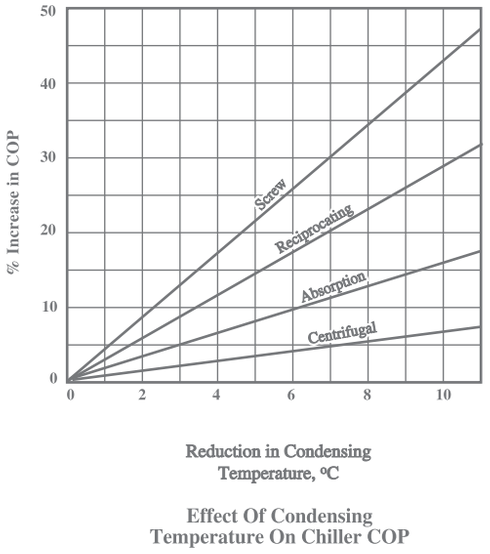


Figure 4.11 Effect of Condensing Temperature on Chiller COP

In the field performance assessment, accurate instruments for inlet and outlet chilled water temperature and condenser water temperature measurement are required, preferably with a least count of 0.1^0C. Flow measurements of chilled water can be made by an ultrasonic flow meter directly or inferred from pump duty parameters. Adequacy check of chilled water is needed often and most units are designed for a typical 0.68 m^3/hr per TR (3 gpm/TR) chilled water flow. Condenser water flow measurement can also be made by a non-contact flow meter directly or inferred from pump duty parameters. Adequacy check of condenser water is also needed often, and most units are designed for a typical 0.91 m^3/hr per TR (4 gpm / TR) condenser water flow.

In case of air conditioning units, the airflow at the Fan Coil Units (FCU) or the Air Handling Units (AHU) can be measured with an anemometer. Dry bulb and wet bulb temperatures are measured at the inlet and outlet of AHU or the FCU and the refrigeration load in TR is assessed as ;

$$TR = \frac{Q \times \rho \times (h_{in} - h_{out})}{3024}$$

Where,

Q is the air flow in m^3/h

ρ is density of air kg/m^3

h_{in} is enthalpy of inlet air kcal/kg

h_{out} is enthalpy of outlet air kcal/kg

Use of psychometric charts can help to calculate h_{in} and h_{out} from dry bulb, wet bulb temperature values which are, in-turn measured, during trials, by a whirling psychrometer.

Power measurements at, compressor, pumps, AHU fans, cooling tower fans can be accomplished by a portable load analyzer.

Estimation of air conditioning load is also possible by calculating various heat loads, sensible and latent based on inlet and outlet air parameters, air ingress factors, air flow, no. of people and type of materials stored.

An indicative TR load profile for air conditioning is presented as follows:

- Small office cabins = 0.1 TR /m^2
- Medium size office i.e., 10 – 30 people occupancy with central A/C = 0.06 TR/ m^2
- Large multistoried office complexes with central A/C = 0.04 TR/ m^2

Integrated Part Load Value (IPLV)

Although the kW/ TR can serve as an initial reference, it should not be taken as an absolute since this value is derived from 100% of the equipment's capacity level and is based on design conditions that are considered the most critical. These conditions occur may be, for example, during only 1% of the total time the equipment is in operation throughout the year. Consequently, it is essential to have data that reflects how the equipment operates with partial loads or in conditions that demand less than 100% of its capacity. To overcome this, an average of kW/TR with partial loads ie Integrated Part Load Value (IPLV) have to be formulated.

The IPLV is the most appropriate reference, although not considered the best, because it only captures four points within the operational cycle: 100%, 75%, 50% and 25%. Furthermore, it assigns the same weight to each value, and most equipment usually operates at between 50 % and 75% of its capacity. This is why it is so important to prepare specific analysis for each case that addresses the four points already mentioned, as well as developing a profile of the heat exchanger's operations during the year.

4.8 Factors Affecting Performance & Energy Efficiency of Refrigeration Plants

Design of Process Heat Exchangers

There is a tendency of the process group to operate with high safety margins which influences the compressor suction pressure / evaporator set point. For instance, a process cooling requirement of 15^0C would need chilled water at a lower temperature, but the range can vary from 6^0C to say 10^0C. At 10^0C chilled water temperature, the refrigerant side temperature has to be lower, say -5^0C to $+5^0$C.

The refrigerant temperature, again sets the corresponding suction pressure of refrigerant which decides the inlet duty conditions for work of compression of the refrigerant compressor. Having the optimum / minimum driving force (temperature difference) can, thus, help to achieve highest possible suction pressure at the compressor, thereby leading to less energy requirement. This requires proper sizing of heat transfer areas of process heat exchangers and evaporators as well as rationalizing the temperature requirement to highest possible value. A 1^0C raise in evaporator temperature can help to save almost 3 % on power consumption. The TR capacity of the same machine will also increase with the evaporator temperature, as given in Table 4.4.

Table 4.4 Effect of Variation in Evaporator Temperature on Compressor Power Consumption

Evaporator Temperature (^{0}C)	Refrigeration Capacity* (tons)	Specific Power Consumption	Increase in kW/ ton (%)
5.0	67.58	0.81	-
0.0	56.07	0.94	16.0
-5.0	45.98	1.08	33.0
-10.0	37.20	1.25	54.0
-20.0	23.12	1.67	106.0

* *Condenser temperature 40^0C*

Towards rationalizing the heat transfer areas, the heat transfer coefficient on refrigerant side can be considered to range from 1400 – 2800 watts /m^2K.

The refrigerant side heat transfer areas provided are of the order of 0.5 Sqm./TR and above in evaporators.

Condensers in a refrigeration plant are critical equipment that influences the TR capacity and power consumption demands. Given a refrigerant, the condensing temperature and corresponding condenser pressure, depend upon the heat transfer area provided, effectiveness of heat exchange and the type of cooling chosen. A lower condensing temperature, pressure, in best of combinations would mean that the compressor has to work between a lower pressure differential as the discharge pressure is fixed by design and performance of the condenser. The choices of condensers in practice range from air cooled, air cooled with water spray, and heat exchanger cooled. Generously sized shell and tube heat exchangers as condensers, with good cooling tower operations help to operate with low discharge pressure values and the TR capacity of the refrigeration plant also improves. With same refrigerant, R22, a discharge pressure of 15 kg/cm^2 with water cooled shell and tube condenser and 20 kg/cm^2 with air cooled condenser indicate the kind of additional work of compression duty and almost 30 % additional energy consumption required by the plant. One of the best option at design stage would be to select generously sized (0.65 m^2/TR and above) shell and tube condensers with water-cooling as against cheaper alternatives like air cooled condensers or water spray atmospheric condenser units.

The effect of condenser temperature on refrigeration plant energy requirements is given in Table 4.5.

Table 4.5 Effect of Variation in Condenser Temperature on Compressor Power Consumption

Condensing Temperature (^{0}C)	Refrigeration Capacity (tons)	Specific Power Consumption (kW / TR)	Increase in kW/ TR (%)
26.7	31.5	1.17	-
35.0	21.4	1.27	8.5
40.0	20.0	1.41	20.5

* *Reciprocating compressor using R-22 refrigerant. Evaporator temperature.-10^{0}C*

Maintenance of Heat Exchanger Surfaces

After ensuring procurement, effective maintenance holds the key to optimizing power consumption.

Heat transfer can also be improved by ensuring proper separation of the lubricating oil and the refrigerant, timely defrosting of coils, and increasing the velocity of the secondary coolant (air, water, etc.). However, increased velocity results in larger pressure drops in the distribution system and higher power consumption in pumps / fans. Therefore, careful analysis is required to determine the most effective and efficient option.

Fouled condenser tubes force the compressor to work harder to attain the desired capacity. For example, a 0.8 mm scale build-up on condenser tubes can increase energy consumption by as much as 35 %. Similarly, fouled evaporators (due to residual lubricating oil or infiltration of air) result in increased power consumption. Equally important is proper selection, sizing, and maintenance of cooling towers. A reduction of 0.55^{0}C temperature in water returning from the cooling tower reduces compressor power consumption by 3.0 % (see Table 4.6).

Table 4.6 Effect of Poor Maintenance on Compressor Power Consumption

Condition	Evaporator Temp. (^{0}C)	Condenser Temp. (^{0}C)	Refrigeration Capacity* (tons)	Specific Power Consumption (kW/ton)	Increase in kW/ Ton (%)
Normal	7.2	40.5	17.0	0.69	-
Dirty condenser	7.2	46.1	15.6	0.84	20.4
Dirty evaporator	1.7	40.5	13.8	0.82	18.3
Dirty condenser and evaporator	1.7	46.1	12.7	0.96	38.7

* *15 ton reciprocating compressor based system. The power consumption is lower than that for systems typically available in India. However, the percentage change in power consumption is indicative of the effect of poor maintenance.*

Multi-Staging for Efficiency

Efficient compressor operation requires that the compression ratio be kept low, to reduce discharge pressure and temperature. For low temperature applications involving high compression ratios, and for wide temperature requirements, it is preferable (due to equipment design limitations) and often economical to employ multi-stage reciprocating machines or centrifugal / screw compressors.

Multi-staging systems are of two-types: compound and cascade – and are applicable to all types of compressors. With reciprocating or rotary compressors, two-stage compressors are preferable for load temperatures from –20 to –58^{0}C, and with centrifugal machines for temperatures around –43^{0}C.

In multi-stage operation, a first-stage compressor, sized to meet the cooling load, feeds into the suction of a second-stage compressor after inter-cooling of the gas. A part of the high-pressure liquid from the condenser is flashed and used for liquid sub-cooling. The second compressor, therefore, has to meet the load of the evaporator and the flash gas. A single refrigerant is used in the system, and the work of compression is shared equally by the two compressors. Therefore, two compressors with low compression ratios can in combination provide a high compression ratio.

For temperatures in the range of –46^{0}C to –101^{0}C, cascaded systems are preferable. In this system, two separate systems using different refrigerants are connected such that one provides the means of heat rejection to the other. The chief advantage of this system is that a low temperature refrigerant which has a high suction temperature and low specific volume can be selected for the low-stage to meet very low temperature requirements.

Matching Capacity to System Load

During part-load operation, the evaporator temperature rises and the condenser temperature falls, effectively increasing the COP. But at the same time, deviation from the design operation point and the fact that mechanical losses form a greater proportion of the total power negate the effect of improved COP, resulting in lower part-load efficiency.

Therefore, consideration of part-load operation is important, because most refrigeration applications have varying loads. The load may vary due to variations in temperature and process cooling needs. Matching refrigeration capacity to the load is a difficult exercise, requiring knowledge of compressor performance, and variations in ambient conditions, and detailed knowledge of the cooling load.

Capacity Control and Energy Efficiency

The capacity of compressors is controlled in a number of ways. Capacity control of reciprocating compressors through cylinder unloading results in incremental (step-by-step) modulation as against continuous capacity modulation of centrifugal through vane control and screw compressors through sliding valves. Therefore, temperature control requires careful system design. Usually, when using reciprocating compressors in applications with widely varying loads, it is desirable to control the compressor by monitoring the return water (or other secondary coolant) temperature rather than the temperature of the water leaving the chiller. This prevents excessive on-off cycling or unnecessary loading / unloading of the compressor. However, if load fluctuations are not high, the temperature of

the water leaving the chiller should be monitored. This has the advantage of preventing operation at very low water temperatures, especially when flow reduces at low loads. The leaving water temperature should be monitored for centrifugal and screw chillers.

Capacity regulation through speed control is the most efficient option. However, when employing speed control for reciprocating compressors, it should be ensured that the lubrication system is not affected. In the case of centrifugal compressors, it is usually desirable to restrict speed control to about 50 % of the capacity to prevent surging. Below 50 %, vane control or hot gas bypass can be used for capacity modulation.

The efficiency of screw compressors operating at part load is generally higher than either centrifugal compressors or reciprocating compressors, which may make them attractive in situations where part-load operation is common. Screw compressor performance can be optimized by changing the volume ratio. In some cases, this may result in higher full-load efficiencies as compared to reciprocating and centrifugal compressors. Also, the ability of screw compressors to tolerate oil and liquid refrigerant slugs makes them preferred in some situations.

Multi-level Refrigeration for Plant Needs

The selection of refrigeration systems also depends on the range of temperatures required in the plant. For diverse applications requiring a wide range of temperatures, it is generally more economical to provide several packaged units (several units distributed throughout the plant) instead of one large central plant. Another advantage would be the flexibility and reliability accorded. The selection of packaged units could also be made depending on the distance at which cooling loads need to be met. Packaged units at load centers reduce distribution losses in the system. Despite the advantages of packaged units, central plants generally have lower power consumption since at reduced loads power consumption can reduce significantly due to the large condenser and evaporator surfaces.

Many industries use a bank of compressors at a central location to meet the load. Usually the chillers feed into a common header from which branch lines are taken to different locations in the plant. In such situations, operation at part-load requires extreme care. For efficient operation, the cooling load, and the load on each chiller must be monitored closely. It is more efficient to operate a single chiller at full load than to operate two chillers at part-load. The distribution system should be designed such that individual chillers can feed all branch lines. Isolation valves must be provided to ensure that chilled water (or other coolant) does not flow through chillers not in operation. Valves should also be provided on branch lines to isolate sections where cooling is not required. This reduces pressure drops in the system and reduces power consumption in the pumping system. Individual compressors should be loaded to their full capacity before operating the second compressor. In some cases it is economical to provide a separate smaller capacity chiller, which can be operated on an on-off control to meet peak demands, with larger chillers meeting the base load.

Flow control is also commonly used to meet varying demands. In such cases the savings in pumping at reduced flow should be weighed against the reduced heat transfer in coils due to reduced velocity. In some cases, operation at normal flow rates, with subsequent longer periods of no-load (or shut-off) operation of the compressor, may result in larger savings.

Chilled Water Storage

Depending on the nature of the load, it is economical to provide a chilled water storage facility with very good cold insulation. Also, the storage facility can be fully filled to meet the process requirements so that chillers need not be operated continuously. This system is usually economical if small variations in temperature are acceptable. This system has the added advantage of allowing the chillers to be operated at periods of low electricity demand to reduce peak demand charges - Low tariffs offered by some electric utilities for operation at night time can also be taken advantage of by using a storage facility. An added benefit is that lower ambient temperature at night lowers condenser temperature and thereby increases the COP.

If temperature variations cannot be tolerated, it may not be economical to provide a storage facility since the secondary coolant would have to be stored at a temperature much lower than required to provide for heat gain. The additional cost of cooling to a lower temperature may offset the benefits. The solutions are case specific. For example, in some cases it may be possible to employ large heat exchangers, at a lower cost burden than low temperature chiller operation, to take advantage of the storage facility even when temperature variations are not acceptable. Ice bank systems which store ice rather than water are often economical.

System Design Features

In overall plant design, adoption of good practices improves the energy efficiency significantly. Some areas for consideration are:

- Design of cooling towers with FRP impellers and film fills, PVC drift eliminators, etc.
- Use of softened water for condensers in place of raw water.
- Use of economic insulation thickness on cold lines, heat exchangers, considering cost of heat gains and adopting practices like infrared thermography for monitoring - applicable especially in large chemical / fertilizer / process industry.
- Adoption of roof coatings / cooling systems, false ceilings / as applicable, to minimize refrigeration load.
- Adoption of energy efficient heat recovery devices like air to air heat exchangers to pre-cool the fresh air by indirect heat exchange; control of relative humidity through indirect heat exchange rather than use of duct heaters after chilling.
- Adopting of variable air volume systems; adopting of sun film application for heat reflection; optimizing lighting loads in the air conditioned areas; optimizing number of air changes in the air conditioned areas are few other examples.

4.9 Performance Assessment of window, split and package air conditioning units

Air Conditioners

Energy Efficiency Ratio (EER): EER is calculated by dividing a chiller's cooling capacity (in watts) by its power input (in watts) at full-load conditions. This definition of EER has been adopted in BEE star labeling programme.

The energy efficiency ratio (EER) = Refrigeration effect in Watts/ Power input in Watts

Based on the condition of the air, the air properties such as specific volume and enthalpy at both inlet and out let conditions can be obtained from psychrometric charts. From these parameters the capacity delivered by the air conditioner can be evaluated, which when compared with power drawn would reveal its performance in terms of kW/TR and EER.

Performance Assessment of Package Air Conditioner – Example

Package air conditioner capacity	-	10 TR
Average Air Velocity	-	2.27 m/s (across suction side filter)
Cross Sectional Area	-	0.58 m^2
Air Flow Rate	-	1.32 m^3/sec - 4751 m^3/hr
Inlet Air Condition		
DBT	-	20 ^oC
WBT	-	14 ^oC
Sp.Vol	-	0.8405 m^3/kg
Enthalpy	-	9.37 kcal/kg
Outlet Air Condition		
DBT	-	12.7 ^oC
WBT	-	11.3 ^oC
Enthalpy	-	7.45 kcal/kg
Cooling Effect Delivered	-	3.6 TR = 12.7 kW
Power Drawn		
Compressor	-	4.71 kW
	-	4.3 kW (shaft power@ 90% motor efficiency)
Pump	-	2.14 kW
C.T Fan	-	0.384 kW
Specific Energy Consumption		
Compressor	-	1.31 kW/TR
Overall	-	2 kW/TR
EER	-	12700/4300
	-	2.95 W/W

Note: The Package A/C unit has two compressors of 5 TR capacity each, of which only one was in operation due to low cooling load.

4.10 Cold Storage Systems

A Refrigerated storage which includes cold storage and frozen food storage is the best known method of preservation of food to retain its value and flavor.

The refrigeration system in a cold storage is usually a vapour compression system comprising the compressor, condenser, receiver, air cooling units and associate piping and controls.

In smaller cold rooms and walk-ins the practice is to use air cooled condensing units with sealed, semi-sealed or open type compressors. In the light of the CFC phase out the trend now is to use HCFC-22, HFC-134a or other substitute refrigerants. In the medium and large sized units the practice is to use a central plant with ammonia as the refrigerant.

In some present day medium and large sized units with pre-fabricated (insulated) panel construction the trend is to use modular HCFC-22/HFC units which are compact, light weight and easy to maintain.

Energy Saving Opportunities in Cold Storage Systems:

Energy cost constitutes a major part of the running cost of a cold store. Apart from the problems of the availability of electrical energy, the ever increasing rate of electrical energy seriously affects the economic viability of cold store units.

Following are some of the measures adopted to achieve energy efficient operation.

- Cold Store Building Design: Proper orientation, compact arrangement of chambers, shading of exposed walls, adequate insulation etc. are some of the important factors.
- Refrigeration System: The system must be designed for optimum operating conditions like evaporating and condensing temperatures, as these conditions have a direct bearing on energy consumption.
- Compressor capacity control system helps in energy savings during partial load operation.
- Control System: The proper control systems for refrigerant level, room temperature, compressor capacity etc., are required to further optimize energy consumption.
- Air Curtain or Strip Curtain: The use of air curtains and strip curtains is a common feature in present day cold stores as they help reduce air infiltration due to frequent and some times long door openings. Fan operated air curtains are expensive and work on electrical power whereas strip curtains are cheaper and need no energy for operation.
- Heat Recovery System: In processing plant cold stores, a heat reclaim system can be installed to recover a part of the heat rejected by the refrigeration system. This can be gainfully utilised in generating hot water free of cost.

4.11 Heat Pumps and Their Applications

Heat Pump Technology

A heat pump is same as an air conditioner except that the heat rejected in an air conditioner becomes the useful heat output. Heat flows naturally from a higher to a lower temperature. Heat pumps, however, are able to force the heat flow in the other direction, using a relatively small amount of high quality

drive energy (electricity, fuel, or high-temperature waste heat). For the example shown in 4.12 the heat pump takes three units of energy from atmosphere and with an additional one unit by way of compressor work is able to provide four units of energy at a higher temperature. Thus heat pumps can transfer heat from natural heat sources in the surroundings, such as the air, ground or water, or from man-made heat sources such as industrial or domestic waste, to a building or an industrial application.

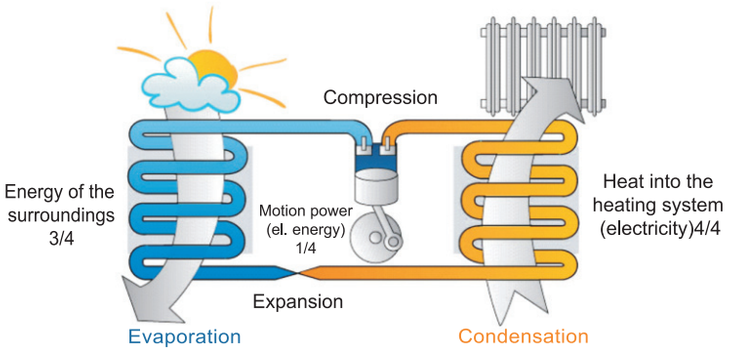


Figure 4.12 Heat Pump

In order to transport heat from a heat source to a heat sink, external energy is needed to drive the heat pump. Theoretically, the total heat delivered by the heat pump is equal to the heat extracted from the heat source, plus the amount of drive energy supplied. Electrically-driven heat pumps for heating buildings typically supply 100 kWh of heat with just 20-40 kWh of electricity. Many industrial heat pumps can achieve even higher performance, and supply the same amount of heat with only 3-10 kWh of electricity. The principle of operation of heat pump is shown in the Figure 4.13.

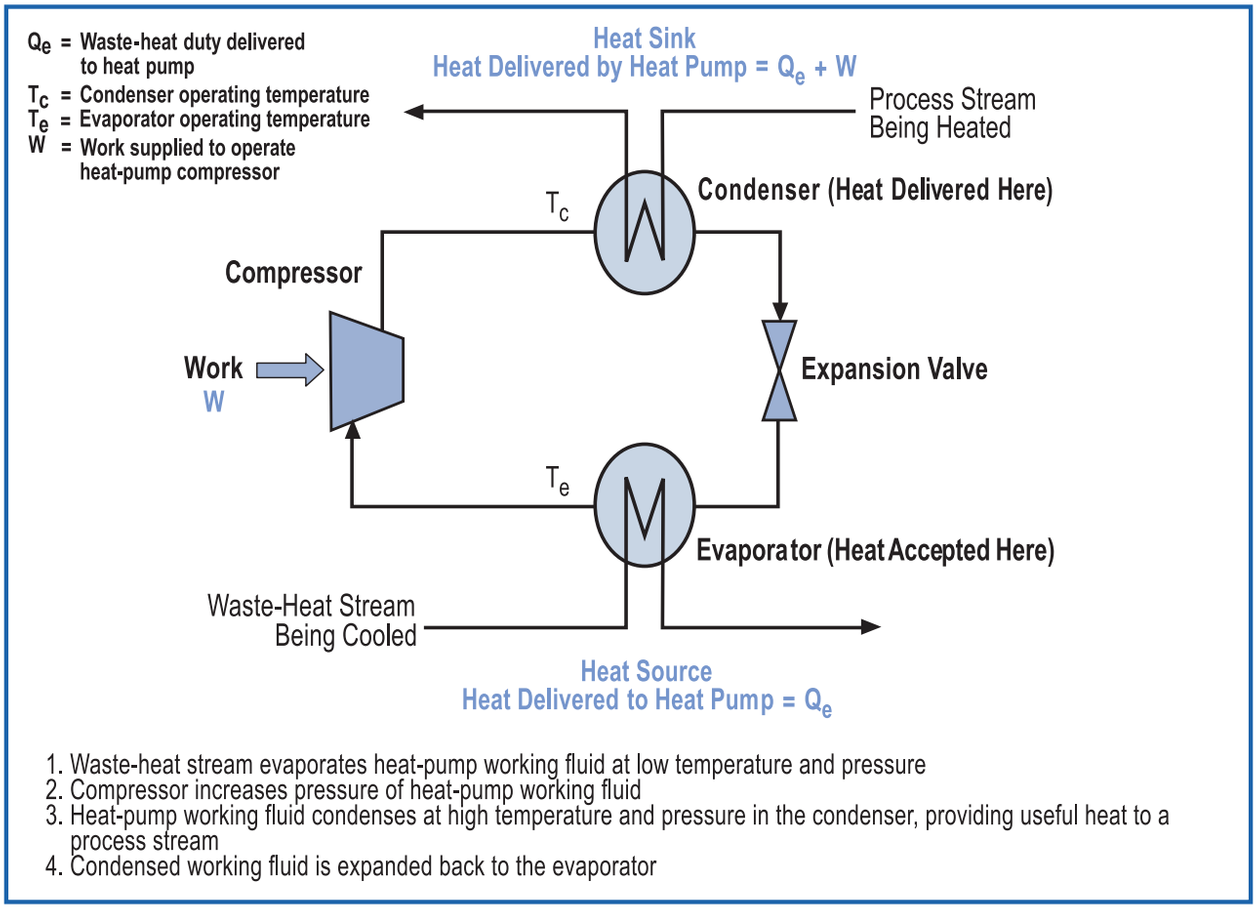


Figure 4.13 Principle of Operation

Heat Pump Applications

Industrial heat pumps are mainly used for:

- Space heating
- Heating of process streams
- Water heating for washing, sanitation and cleaning
- Steam production
- Drying/dehumidification
- Evaporation
- Distillation
- Concentration

When heat pumps are used in drying, evaporation and distillation processes, heat is recycled within the process. For heating of the space, process streams and steam production, the heat pumps utilise (waste) heat sources between 20ºC and 100ºC.

4.12 Ventilation Systems

Ventilation can be simply described as air circulation, the extraction of stale, overheated and contaminated air and supply and distribution of fresh air in amounts necessary to provide healthy and comfortable conditions for the occupants of the room. The ventilation effectiveness is dictated by number of Air Changes per Hour (ACH). The number air changes depend on the purpose and function. Typical air changes are given in Table 4.7 for various operations.

Table 4.7 Typical Air Changes per Hour

Location	Air changes per hour
Boiler room	15 – 30
Compressor room	10 – 12
Conference rooms	10 – 20
Engine rooms	15 – 30
Lavatories	6 – 15
Offices	6 – 10
Welding shops	15 – 30

Calculation of ventilation rate: If the compressor room size is 15 m (L) X 10 m (B) X 4 m (H) then the ventilation rate is

= L x B X H X ACH

= 15 X 10 X 4 X 10

= 6000 m^3/hr

4.13 Ice Bank Systems

Ice Bank System is a proven technology that has been utilized for decades Thermal energy storage takes advantage of low cost, off-peak electricity, produced more efficiently throughout the night, to create and store cooling energy for use when electricity tariffs are higher, typically during the day. There are full- and partial- load Off-Peak Cooling systems. The essential element for either full- or partial- storage configurations are thermal- energy storage tanks. Each tank contains a spiral-wound, polyethylene-tube heat exchanger surrounded with water. ICEBANK tanks are available in a variety of sizes ranging from 45 to over 500 ton-hours. These systems are economical based on the electricity tariff of particular utility. These systems can be employed to meet the air conditioning requirements in the commercial buildings as well as to meet the chilling requirements in Diary and process industry. The main advantage of these systems is to reduce the peak demand of the utility and also reduce the cost of operation for the end user

How Ice Bank Works?

With a partial-storage system, the chiller can be 40 to 50 percent smaller than other HVAC systems, because the chiller works in conjunction with the ICEBANK tanks during on-peak daytime hours to manage the building's cooling load. During off-peak night time hours, the chiller charges the ICEBANK tanks for use during the next day's cooling. The lowest possible average load is obtained by extending the chiller hours of operation.

4.14 Humidification Systems

This is a process involving reduction in dry bulb temperature and increase in specific humidity. The atmospheric conditions with respect to humidity play a very important part in many manufacturing processes. For example in textile processing the properties like **dimensions, weight, tensile** strength, elastic recovery, electrical resistance, rigidity etc. of all textile fibre are influenced by humidity maintained. Temperature does not have a great effect on the fibres but the temperature dictates the amount of moisture the air will hold in suspension and, therefore, temperature and humidity must be considered together. Humidification system without chilling helps to maintain only the RH% without much difficulty.

Adiabatic saturation or evaporative cooling

In this process (Figure 4.14) air comes in direct contact with water in the air washer. There is heat and mass transfer between air and water. The humidity ratio of air increases. If the time of contact is sufficient, the air gets saturated. Latent heat of evaporation required for conversion of water into water vapor is taken from the remaining water. When equilibrium conditions are reached, water cools down to the wet bulb temperature of the air. If the air washer is ideal, the dry bulb temperature and wet bulb temperature of the air would be equal. Dry bulb temperature of the air goes down in the process and the effect of cooling is due to the evaporation of some part of the water. That is why it is called **Evaporative Cooling.**

The sensible heat is decreased as the temperature goes down but the latent heat goes up as water vapour is added to the air. The latent heat required by the water which is evaporated in the air is drawn from

the sensible heat of the same air. Thus it is transformation of sensible heat to latent heat. During this process the enthalpy of air remains the same. If humidity ratios of saturated air and of the air before saturation are known, then the difference between the two would be the amount of water vapour absorbed by unit weight of dry air.

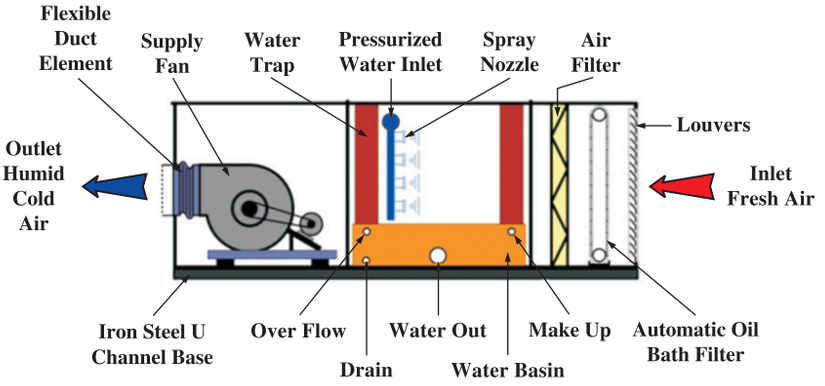


Figure 4.14 Air Washer Unit

Humidifying Air by adding Water

If water is added to air without any heat supply the state of air change **adiabatic** along a constant enthalpy line in the *psychrometric chart*. The dry bulb temperature of the air decreases.

The amount of added water can be expressed as

$$m_w = v \rho (\omega out - \omega_{in})$$

Where,

m_w = mass of added water (kg/hr)

v = volume flow of air (m^3/hr)

ρ = density of air - vary with temperature, 1.293 kg/m^3 at 20°C (kg/m^3)

ω = specific humidity of air (kg/kg)

Example - Humidifying Air by adding Water

In an air washer of textile humidification system airflow of 3000 m^3/h at 25°C and 10% relative humidity is humidified to 60% relative humidity by adding water through spray nozzles. Calculate the amount of water required. The specific humidity of air at inlet and outlet are 0.002 kg/kg and 0.0062 kg/kg respectively.

Solution

The amount of water added can be calculated as:

m_w = 3000 X 1.184 X (0.0062 0.002)

= 14.9 kg/h

4.15 Standards and Labeling of Room Air Conditioners

This Standard specifies the energy labeling requirements for single-phase split and unitary air conditioners of the vapour compression type for household use up to a rated cooling capacity of 11 kW and that fall within the scope of IS1391 Part 1 and Part 2, being manufactured, imported, or sold in India.

The number of stars displayed on the energy label. The available stars are between a minimum of one and a maximum of five shown in one star interval. The star rating is calculated from the Star Rating Band.

The Table 4.8 & 4.9 below is valid for split and unitary type air conditioners between 01 January 2014 and 31st December 2015.

Table 4.8 Star Rating Band valid for Split type air conditioners (from 01 January 2014 to 31 December 2015)

Star Rating	Energy Efficiency Ratio (Watt/Watt)	
	Minimum	Maximum
1 Star *	2.70	2.89
2 Star * *	2.90	3.09
3 Star * * *	3.10	3.29
4 Star * * * *	3.30	3.49
5 Star * * * * *	3.50	

Table 4.9 Star Rating Band valid for Unitary type air conditioners (from 01 January 2014 to 31 December 2015)

Star Rating	Energy Efficiency Ratio (Watt/Watt)	
	Minimum	Maximum
1 Star *	2.50	2.69
2 Star * *	2.70	2.89
3 Star * * *	2.90	3.09
4 Star * * * *	3.10	3.29
5 Star * * * * *	3.30	

4.16 Energy Saving Opportunities

a) Cold Insulation

Insulate all cold lines / vessels using economic insulation thickness to minimize heat gains; and choose appropriate (correct) insulation.

b) Building Envelope

Optimise air conditioning volumes by measures such as use of false ceiling and segregation of critical areas for air conditioning by air curtains.

c) Building Heat Loads Minimisation

Minimise the air conditioning loads by measures such as roof cooling, roof painting, efficient lighting, pre-cooling of fresh air by air- to-air heat exchangers, variable volume air system, otpimal thermo-static setting of temperature of air conditioned spaces, sun film applications, etc.

d) Process Heat Loads Minimisation

Minimize process heat loads in terms of TR capacity as well as refrigeration level, i.e., temperature required, by way of:

i) Flow optimization
ii) Heat transfer area increase to accept higher temperature coolant
iii) Avoiding wastages like heat gains, loss of chilled water, idle flows.
iv) Frequent cleaning / de-scaling of all heat exchangers

f) At the Refrigeration A/C Plant Area

i) Ensure regular maintenance of all A/C plant components as per manufacturer guidelines.
ii) Ensure adequate quantity of chilled water and cooling water flows, avoid bypass flows by closing valves of idle equipment.
iii) Minimize part load operations by matching loads and plant capacity on line; adopt variable speed drives for varying process load.
iv) Make efforts to continuously optimize condenser and evaporator parameters for minimizing specific energy consumption and maximizing capacity.
v) Adopt VAR system where economics permit as a non-CFC solution.

4.17 Case Study: Screw Compressor Application

Background

Rotary Screw Compressors are widely used for refrigeration applications to compress ammonia & other refrigerating gases. A typical sectional view of the compressor is shown below in Figure 4.15.

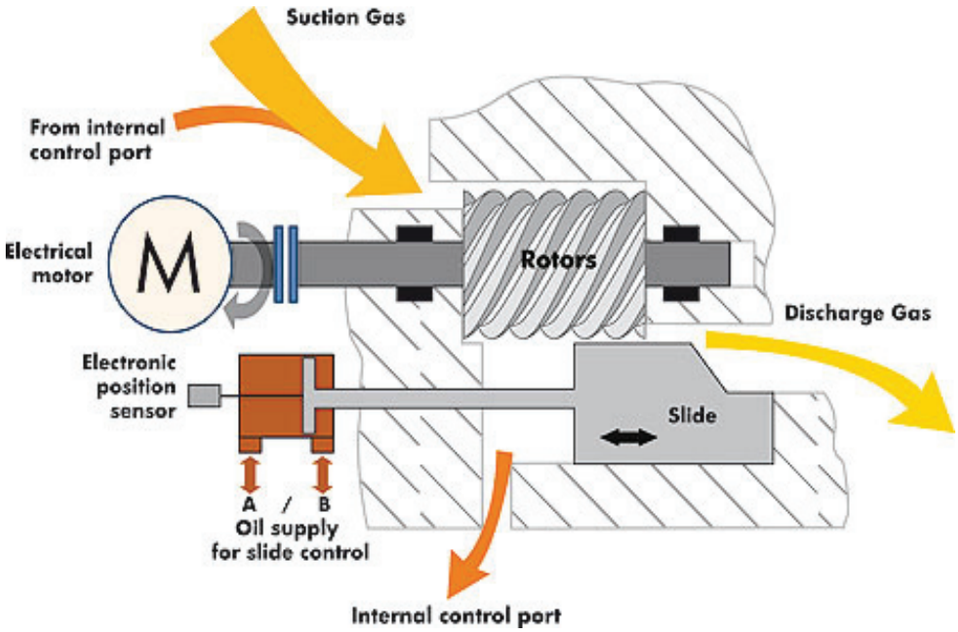


Figure 4.15 Section of Screw Compressor showing the meshing screws & slide valve

Capacity control is used in Screw Compressors to vary the amount of gas drawn into the compressor. This is necessary to provide accurate suction pressure control as the evaporator load varies. One of the commonly used methods used is slide valve control of discharge port. It enables capacity control from 10-100%.

VFD's are increasingly used for capacity control of screw compressors as they are more energy efficient and can be used in a closed loop control system for precise control of suction pressure as evaporator load varies.

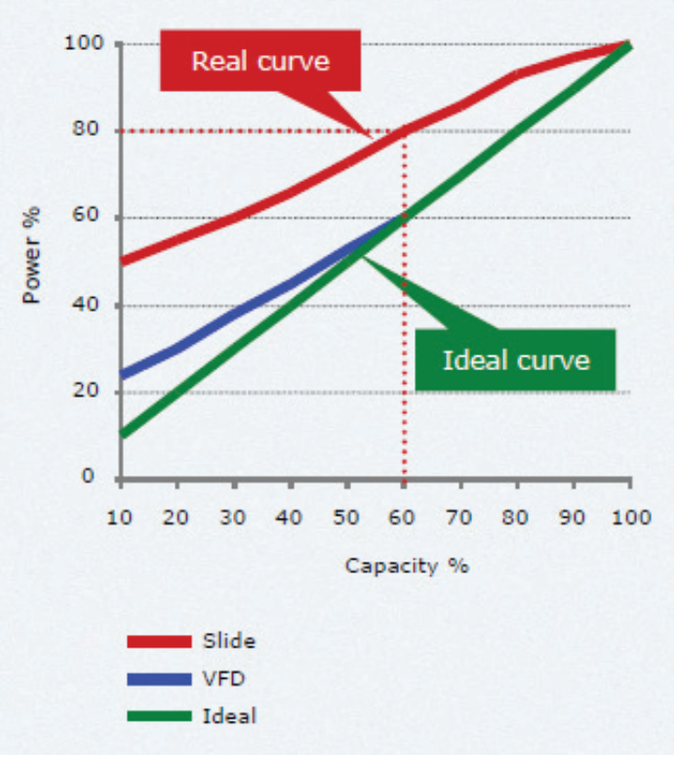
Use of VFD for screw compressor motor in brewery plant

The screw compressor is part of the refrigeration system which uses ammonia gas as the refrigerant for producing chilled water. The drive motor for the compressor has the following nameplate data:

1. Capacity -315kW.
2. Supply Voltage - 415V, 3Φ, 50Hz.
3. Design Speed - 3000RPM.
4. Full Load Current (FLC) - 620A.

Trials were conducted with a VFD suitable for the above-mentioned motor. For the purpose of comparison, readings were taken for both cases-compressor running with slide valve capacity control, and control with VFD.

Figure 4.16 below shows the typical power consumption variation with load with and without VFD.



Note:

- Slide valve position is determined by suction pressure PID control.
- However, the response to load changes is delayed due to a dead band of 300seconds.
- This fact results in increased energy losses and reduces the COP of the compressor.
- Typically, 60% capacity requires 80% power.

Figure 4.16 Comparisons of power consumption data for screw compressor (with & without VFD)

The table 4.10 below gives the measured system data during compressor running at part load condition.

Table 4.10 Performance parameters for partial load conditions

Volumetric refrigerant flow (m^3)	Suction Pressure (kg/cm^2)	Discharge Pressure (kg/cm^2)	Inlet Temp. (°C)	Outlet Temp. (°C)	Total refrigeration load (Calories)	Total refrigeration load (TR)
21.03	3.19	11.89	28.08	1.49	558964	184.73
26.62	3.16	11.65	26.85	2.12	658336	217.57
26	3.17	11.63	27.75	1.68	678000	224.07
28	3.22	11.7	27.91	1.92	727650	240.47
32	3.22	11.78	27.98	2.28	822400	271.79
20.13	3.25	12.22	27.28	1.5	518947	171.5
21.71	3.25	12.38	27.77	1.6	568197	187.78
23.36	3.25	12.52	28.4	1.64	625306	206.65
28.15	3.25	11.99	28.58	1.27	714310	236.07
28.51	3.25	12.31	28.7	1.54	774362	255.91

The table 4.11 below gives the power consumption data for partial load operation with and without VFD.

Table 4.11 Power consumption for partial load operation (with & without VFD)

Power consumed without VFD (kW)	Power consumed with VFD (kW)	Measured Power consumed per ton of refrigeration load without VFD (kW/TR)	Measured Power consumed per ton of refrigeration load with VFD (kW/TR)	Hourly savings due to VFD operation (kW)
173.14	127.25	0.94	0.69	45.89
203.93	149.87	0.94	0.69	54.06
210.02	154.35	0.94	0.69	55.67
225.4	165.65	0.94	0.69	59.75
254.75	187.22	0.94	0.69	67.53
160.75	118.14	0.94	0.69	42.61
176	129.35	0.94	0.69	46.65
193.69	142.35	0.94	0.69	51.34
221.26	162.61	0.94	0.69	58.65
239.87	176.28	0.94	0.69	63.59

Average hourly power savings = 57.6 kW

Average yearly energy savings (250 days x 10 hours) = 136435 kWh

Average yearly monetary savings (Rs. 6.00/kWh) = Rs. 8,18,610 /-

Solved Example:

The measured values of a 20 TR package air conditioning plant are given below:

Average air velocity across suction side filter: 2.5 m/s

Cross Sectional area of suction: 1.2 m^2

Inlet air = Dry Bulb:20^oC, Wet Bulb: 14 ^oC, Enthalpy: 9.37 kcal/kg

Outlet air = Dry Bulb: 12.7 ^oC, Wet Bulb: 11.3 ^oC; Enthalpy: 7.45 kcal/kg

Specific volume of air: 0.85 m^3/kg

Power drawn: by compressor: 10.69 kW

by Pump: 4.86 kW

by Cooling tower fan: 0.87 kW

Calculate:

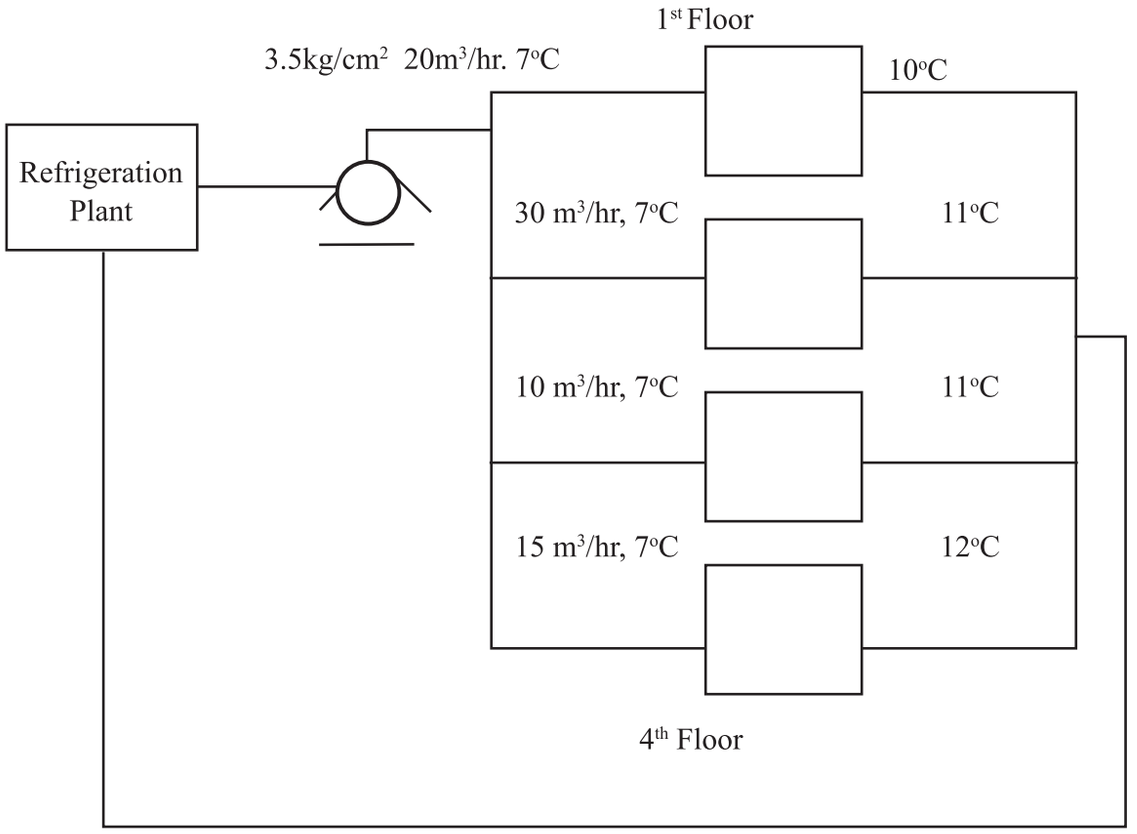
a) Air Flow rate in m^3/hr
b) Cooling effect delivered in kW
c) Specific power consumption of compressor in kW/TR
d) Overall kW/TR
e) Energy Efficiency Ratio in kW/kW

Ans:

a)	Air flow rate = 2.5*1.2	=	3 m^3/sec = 10800 m^3/hr
b)	Cooling Effect delivered	=	[(9.37-7.45)*10800]/(0.85*3024)
		=	8.07 TR = 28.32 kW
c)	Compressor kW/TR = 10.69/8.07	=	1.32
d)	Overall kW/TR = (10.69+4.86+0.87)/8.07	=	2.04
e)	Energy Efficiency Ratio(EER) in kW/kW	=	28.32/10.69 = 2.65

QUESTIONS	
Objective Type Questions	
1.	Higher COP can be achieved with_____. a) lower evaporator temperature and higher condenser temperature b) higher evaporator temperature and lower condenser temperature c) higher evaporator temperature and higher condenser temperature d) lower evaporator temperature and lower condenser temperature
2.	Li – Br water absorption refrigeration systems have a COP in the range of a) 0.4 – 0.5 b) 0.65 – 0.70 c) 0.75 – 0.80 d) none of the above
3.	Hermetic system are used in a) domestic refrigerators b) centrifugal chillers c) screw chillers d) large reciprocating chillers
4.	Which of the following can be used as refrigeration both in vapour compressor and vapour absorption system a) ammonia b) R – 11 c) R-12 d) lithium Bromide
5.	One ton of refrigeration (TR) is equal to a) 3.51 kW b) 3024 kcal/hr c) 12,000 BTU/hr d) all of the above
6.	The refrigeration load in TR when 20 m^3/hr of water is cooled from 17° C to 10° C is about a) 46.3 b) 41.6 c) 116.6 d) none of the above
7.	At dew point temperature moisture in air a) increases b) decreases c) remains same d) begins to condense
8.	Which of the following is not a part of the vapour compression refrigeration system a) compressor b) evaporator c) condenser d) absorber
9.	The COP of a vapour compression refrigeration system is 3.0. If the power input to compressor is 100 kW , the tonnage of refrigeration system is given by a) 85.3 b) 9.48 c) 300 d) none of the above
10.	The term Refrigeration means a) addition of cooling b) removal of heat c) removal and relocation of heat d) replacement of heat
Short Type Questions	
S-1	Discuss in brief any three methods by which energy can be saved in an air conditioning system.
S-2	What are the parameters required to be measured while estimating the chiller performance in kW/TR?
S-3	Explain the term Integrated Part Load Value (IPLV)?
S-4	Name five parameters that a psychometric chart provides for an air conditioning engineer?
S-5	Define one ‘Ton of Refrigeration (TR)’.

	Long Type Questions
L-1	The management of a building facility retains the services of an energy auditor to asses the refrigeration load of each floor of the building and efficiency of the chilled water pump. The auditor measures the chilled water flow of each AHU by non contact flow meter and measure the temperature at the inlet and outlet of AHUs, as per the details given below. The discharge pressure gauge of the chilled water pump shows 3.5 kg/cm2 and the refrigeration tank water level is 1 metre above the pump centerline. The power drawn by the refrigeration compressor and pump motor was found to be 89kW and 17.58 kW respectively.



What would be the analysis of the auditor on the following?

a) Floor wise refrigeration load requirement (in TR)
b) Specific power consumption of Refrigeration compressor (kW/TR) at the motor operating efficiency of 92%
c) Chilled water pump efficiency , at the motor operating efficiency of 89%

L-2	Explain the operating principle of a vapour compression and a vapour absorption refrigeration sytem

REFERENCES

1. Technology Menu on Energy Efficiency (NPC)
2. ASHRAE Hand Book
3. NPC Case Studies
4. Inputs on VFD for screw compressor: Danfoss India Pvt. Ltd.

Chapter 5

Fans & Blowers

Learning Objectives

In this chapter you will learn about

- Fan types and applications
- Fan characteristics and performance curves
- Fan laws
- Factors affecting efficiency
- Design and selection criteria
- Pressure drop and system resistance
- Flow control strategies
- Performance assessment
- Energy saving opportunities

5. FANS AND BLOWERS

Types, Performance evaluation, Efficient system operation, Pressure drop, Flow control strategies and energy conservation opportunities.

5.1 Introduction

Fans and blowers provide air for ventilation and industrial process requirements. Fans generate a pressure to move air (or gases) against a resistance caused by ducts, dampers, or other components in a fan system. The fan rotor receives energy from a rotating shaft and transmits it to the air.

Difference between Fans, Blowers and Compressors

Fans, blowers and compressors are differentiated by the method used to move the air, and by the system pressure they must operate against. As per American Society of Mechanical Engineers (ASME) the specific ratio - the ratio of the discharge pressure over the suction pressure - is used for defining the fans, blowers and compressors (see Table 5.1).

Table 5.1 Differences between Fans, Blower and Compressor

Equipment	Specific Ratio	Pressure rise (mmWg)
Fans	Up to 1.11	1136
Blowers	1.11 to 1.20	1136 – 2066
Compressors	more than 1.20	-

5.2 Fan Types

Fan and blower selection depends on the volume flow rate, pressure, type of material handled, space limitations, and efficiency. Fan efficiencies differ from design to design and also by types. Typical ranges of fan efficiencies are given in Table 5.2.

Fans fall into two general categories: centrifugal flow and axial flow.

In centrifugal flow, airflow changes direction twice - once when entering and second when leaving (forward curved, backward curved or inclined, radial) (see Figure 5.1).

In axial flow, air enters and leaves the fan with no change in direction (propeller, tubeaxial, vaneaxial) (see Figure 5.2).

Table 5.2 Fan Efficiencies

Type of fan	Peak Efficiency Range
Centrifugal Fan	
Airfoil, backward curved/inclined	79-83
Modified radial	72-79
Radial	69-75
Pressure blower	58-68
Forward curved	60-65
Axial fan	
Vanaxial	78-85
Tubeaxial	67-72
Propeller	45-50

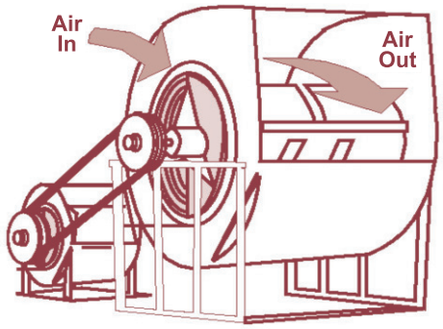


Figure 5.1 Centrifugal Fan

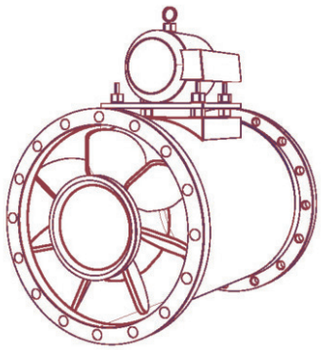


Figure 5.2 Axial Fan

Centrifugal Fan: Types

The major types of centrifugal fan are: *radial, forward curved and backward curved* (see Figure 5.3).

Radial fans are industrial workhorses because of their high static pressures (upto 1400 mm WC) and ability to handle heavily contaminated airstreams. Because of their simple design, radial fans are well suited for high temperatures and medium blade tip speeds.

Forward-curved fans are used in clean environments and operate at lower temperatures. They are well suited for low tip speed and high-airflow work - they are best suited for moving large volumes of air against relatively low pressures.

Backward-inclined fans are more efficient than forward-curved fans. Backward-inclined fans reach their peak power consumption and then power demand drops off well within their useable airflow range. Backward-inclined fans are known as "non-overloading" because changes in static pressure do not overload the motor.

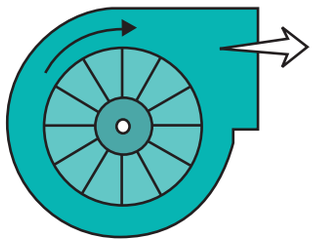
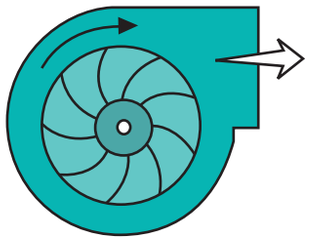
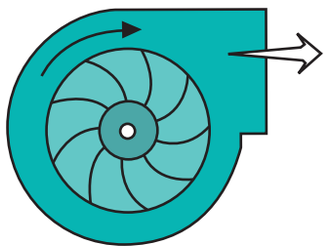
Paddle Blade (Radial blade)	Forward Curved (Multi-Vane)	Backward Curved
		

Figure 5.3 Types of Centrifugal Fans

Axial Flow Fan: Types

The major types of axial flow fans are: *tube axial, vane axial and propeller (see Figure 5.4.)*

Tubeaxial fans have a wheel inside a cylindrical housing, with close clearance between blade and housing to improve airflow efficiency. The wheel turn faster than propeller fans, enabling operation under high-pressures 250 – 400 mm WC. The efficiency is up to 65%.

Vaneaxial fans are similar to tubeaxials, but with addition of guide vanes that improve efficiency by directing and straightening the flow. As a result, they have a higher static pressure with less dependence on the duct static pressure. Such fans are used generally for pressures upto 500 mmWC. Vaneaxials are typically the most energy-efficient fans available and should be used whenever possible.

Propeller fans usually run at low speeds and moderate temperatures. They experience a large change in airflow with small changes in static pressure. They handle large volumes of air at low pressure or free delivery. Propeller fans are often used indoors as exhaust fans. Outdoor applications include air-cooled condensers and cooling towers. Efficiency is low – approximately 50% or less.

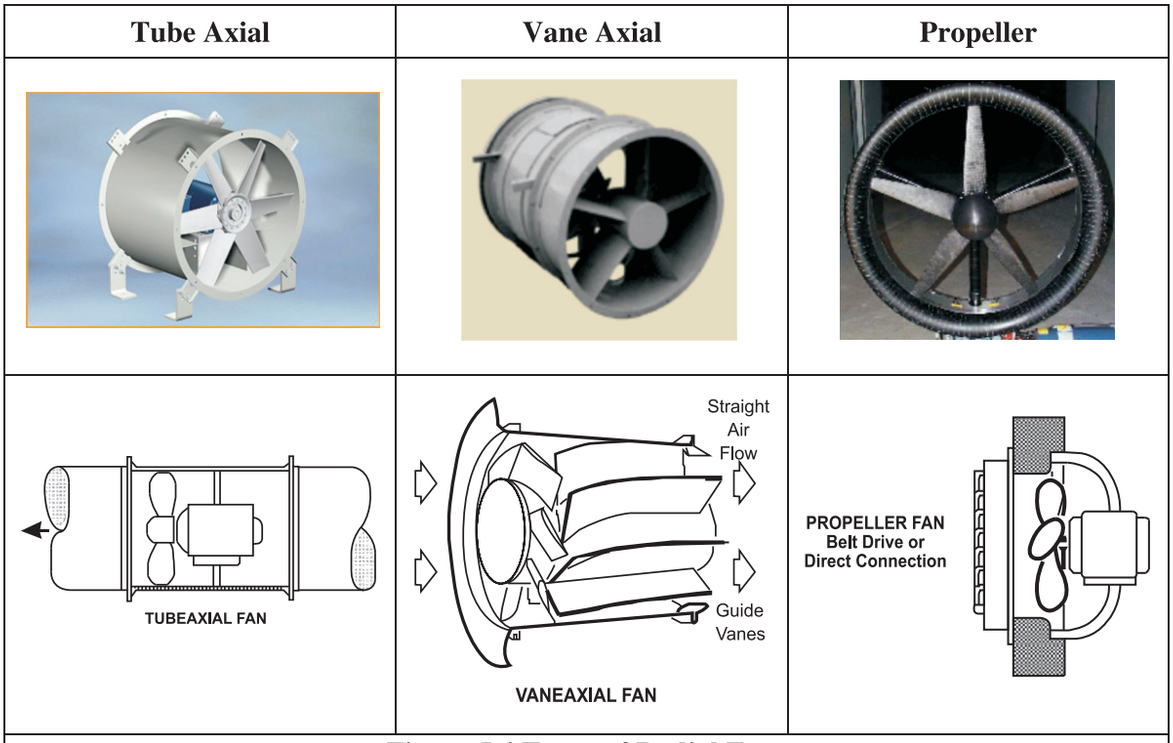


Figure 5.4 Types of Radial Fans

The different types of fans, their characteristics and typical applications are given in Table 5.3.

Common Blower Types

Blowers can achieve much higher pressures than fans, as high as 1.20 kg/cm^2. They are also used to produce negative pressures for industrial vacuum systems. Major types are: centrifugal blower and positive-displacement blower.

Centrifugal blowers look more like centrifugal pumps than fans. The impeller is typically driven and rotates as fast as 15,000 rpm. In multi-stage blowers, air is accelerated as it passes through each impeller. In single-stage blower, air does not take many turns, and hence it is more efficient.

Centrifugal blowers typically operate against pressures of 0.35 to 0.70 kg/cm^2, but can achieve higher pressures. One characteristic is that airflow tends to drop drastically as system pressure increases, which can be a disadvantage in material conveying systems that depend on a steady air volume. Because of this, they are most often used in applications that are not prone to clogging.

Table 5.3 Types of Fans, Characteristics, and Typical Applications					
Centrifugal Fans			Axial-flow Fans		
Type	**Characteristics**	**Typical Applications**	**Type**	**Characteristics**	**Typical Applications**
Radial	High pressure, medium flow, efficiency close to tube-axial fans, power increases continuously	Various industrial applications, suitable for dust laden, moist air/ gases	Propeller	Low pressure, high flow, low efficiency, peak efficiency close to point of free air delivery (zero static pressure)	Air-circulation, ventilation, exhaust
Forward-curved blades	Medium pressure, high flow, dip in pressure curve, efficiency higher than radial fans, power rises continuously	Low pressure HVAC, packaged units, suitable for clean and dust laden air / gases	Tube-axial	Medium pressure, high flow, higher efficiency than propeller type, dip in pressure-flow curve before peak pressure point.	HVAC, drying ovens, exhaust systems
Backward curved blades	High pressure, high flow, high efficiency, power reduces as flow increases beyond point of highest efficiency	HVAC, various industrial applications, forced draft fans, etc.	Vane-axial	High pressure, medium flow, dip in pressure-flow curve, use of guide vanes improves efficiency	High pressure applications including HVAC systems, exhausts
Airfoil type	Same as backward curved type, highest efficiency	Same as backward curved, but for clean air applications			

Positive-displacement blowers have rotors, which "trap" air and push it through housing. Positive-displacement blowers provide a constant volume of air even if the system pressure varies. They are especially suitable for applications prone to clogging, since they can produce enough pressure - typically up to 1.25 kg/cm^2 - to blow clogged materials free. They turn much slower than centrifugal blowers (e.g. 3,600 rpm), and are often belt driven to facilitate speed changes.

5.3 Fan Performance Evaluation and Efficient System Operation

System Characteristics

The term "system resistance" is used when referring to the static pressure. The system resistance is the sum of static pressure losses in the system. The system resistance is a function of the configuration of ducts, pickups, elbows and the pressure drops across equipment-for example bagfilter or cyclone.

The system resistance varies with the square of the volume of air flowing through the system. For a given volume of air, the fan in a system with narrow ducts and multiple short radius elbows is going to have to work harder to overcome a greater system resistance than it would in a system with larger ducts and a minimum number of long radius turns. Long narrow ducts with many bends and twists will require more energy to pull the air through them. Consequently, for a given fan speed, the fan will be able to pull less air through this system than through a short system with no elbows. Thus, the system resistance increases substantially as the volume of air flowing through the system increases; square of air flow.

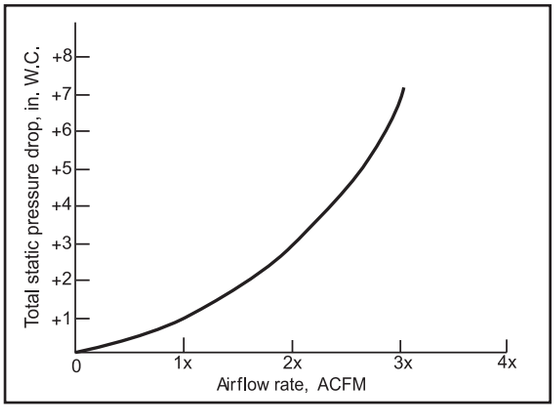


Figure 5.5 System Characteristics

Conversely, resistance decreases as flow decreases. To determine what volume the fan will produce, it is therefore necessary to know the system resistance characteristics.

In existing systems, the system resistance can be measured. In systems that have been designed, but not built, the system resistance must be calculated. Typically a system resistance curve (see Figure 5.5) is generated with for various flow rates on the x-axis and the associated resistance on the y-axis.

Fan Characteristics

Fan characteristics can be represented in form of fan curve(s). The fan curve is a performance curve for the particular fan under a specific set of conditions. The fan curve is a graphical representation of a number of inter-related parameters. Typically a curve will be developed for a given set of conditions usually including: fan volume, system static pressure, fan speed, and brake horsepower required to drive the fan under the stated conditions. Some fan curves will also include an efficiency curve so that a system designer will know where on that curve the fan will be operating under the chosen conditions (see Figure 5.6). In the many curves shown in the Figure, the curve static pressure (SP) vs. flow is especially important.

The intersection of the system curve and the static pressure curve defines the operating point. When the system resistance changes, the operating point is also changes. Once the operating point is fixed, the power required could be found by following a vertical line that passes through the operating point to an intersection with the power (BHP) curve. A horizontal line drawn through the intersection with the power curve will lead to the required power on the right vertical axis. In the depicted curves, the fan efficiency curve is also presented.

System Characteristics and Fan Curves

In any fan system, the resistance to air flow (pressure) increases when the flow of air is increased. As mentioned before, it varies as the square of the flow. The pressure required by a system over a range of flows can be determined and a “system performance curve” can be developed (shown as SC) (see Figure 5.7).

This system curve can then be plotted on the fan curve to show the fan's actual operating point at "A" where the two curves (N_1 and SC_1) intersect. This operating point is at air flow Q_1 delivered against pressure P_1.

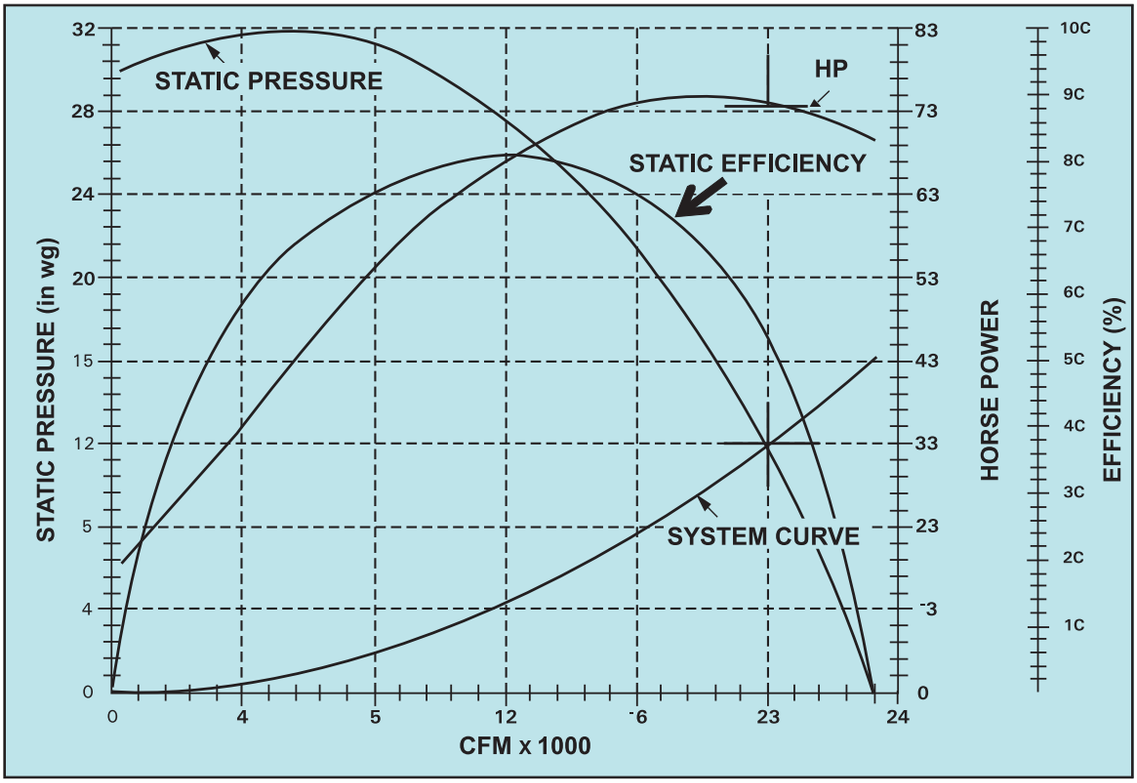


Figure 5.6 Fan Characteristics Curve by Manufacturer

A fan operates along a performance given by the manufacturer for a particular fan speed. (The fan performance chart shows performance curves for a series of fan speeds.) At fan speed N_1, the fan will operate along the N_1 performance curve as shown in Figure 5.7. The fan's actual operating point on this curve will depend on the system resistance; fan's operating point at "A" is flow (Q_1) against pressure (P_1).

Two methods can be used to reduce air flow from Q1 to Q_2:

First method is to restrict the air flow by partially closing a damper in the system. This action causes a new system performance curve (SC_2) where the required pressure is greater for any given air flow. The fan will now operate at "B" to provide the reduced air flow Q_2 against higher pressure P_2.

Second method to reduce air flow is by reducing the speed from N_1 to N_2, keeping the damper fully open. The fan would operate at "C" to provide the same Q_2 air flow, but at a lower pressure P_3.

Thus, reducing the fan speed is a much more efficient method to decrease airflow since less power is required and less energy is consumed.

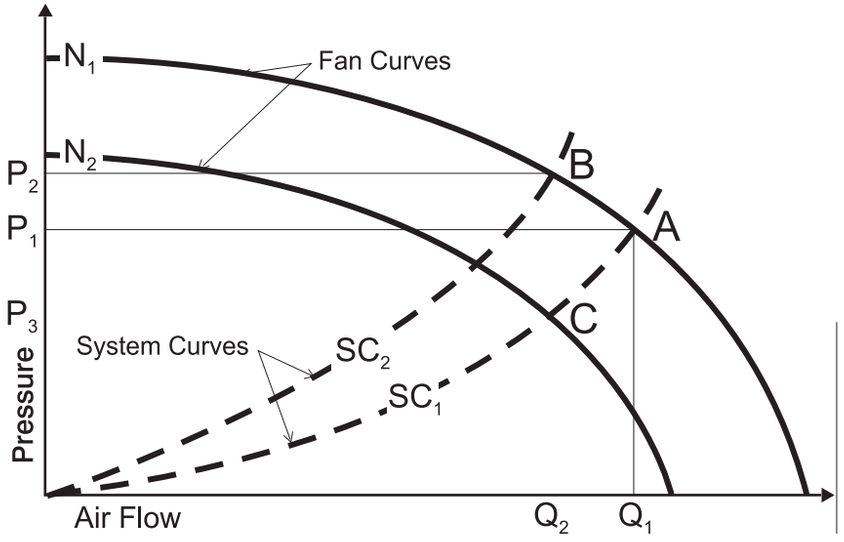
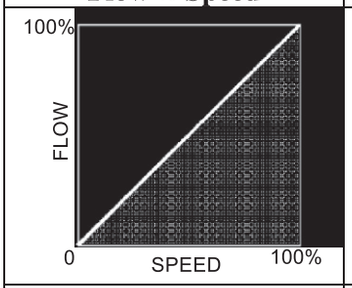
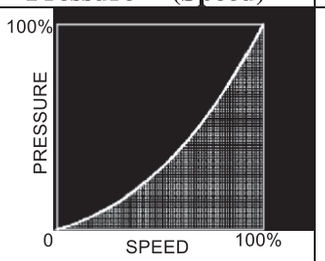
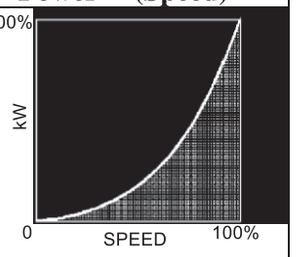


Figure 5.7 System Curve

Fan Laws

The fans operate under a predictable set of laws concerning speed, power and pressure. A change in speed (RPM) of any fan will predictably change the pressure rise and power necessary to operate it at the new RPM.

Flow ∞ Speed	**Pressure ∞ (Speed)^2**	**Power ∞ (Speed)^3**
		
$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$	$\frac{SP_1}{SP_2} = \left(\frac{N_1}{N_2}\right)^2$	$\frac{kW_1}{kW_2} = \left(\frac{N_1}{N_2}\right)^3$
Varying the RPM by 10% decreases or increases air delivery by 10%.	***Reducing the RPM by 10% decreases the static pressure by 19% and an increase in RPM by 10% increases the static pressure by 21%.***	***Reducing the RPM by 10% decreases the power requirement by 27% and an increase in RPM by 10% increases the power requirement by 33%.***

Where Q – Flow, SP – Static Pressure, kW – Power and N – Speed (RPM)

5.4 Fan Design and Selection Criteria

Precise determination of air-flow and required outlet pressure are most important in proper selection of fan type and size. The air-flow required depends on the process requirements; normally determined

from heat transfer rates, or combustion air or flue gas quantity to be handled. System pressure requirement is usually more difficult to compute or predict. Detailed analysis should be carried out to determine pressure drop across the length, bends, contractions and expansions in the ducting system, pressure drop across filters, drop in branch lines, etc. These pressure drops should be added to any fixed pressure required by the process (in the case of ventilation fans there is no fixed pressure requirement). Frequently, a very conservative approach is adopted allocating large safety margins, resulting in over-sized fans which operate at flow rates much below their design values and, consequently, at very poor efficiency.

Once the system flow and pressure requirements are determined, the fan and impeller type are then selected. For best results, values should be obtained from the manufacturer for specific fans and impellers.

The choice of fan type for a given application depends on the magnitudes of required flow and static pressure. For a given fan type, the selection of the appropriate impeller depends additionally on rotational speed. Speed of operation varies with the application. High speed small units are generally more economical because of their higher hydraulic efficiency and relatively low cost. However, at low pressure ratios, large, low-speed units are preferable.

Fan Performance and Efficiency

Typical static pressures and power requirements for different types of fans are given in the Figure 5.8.

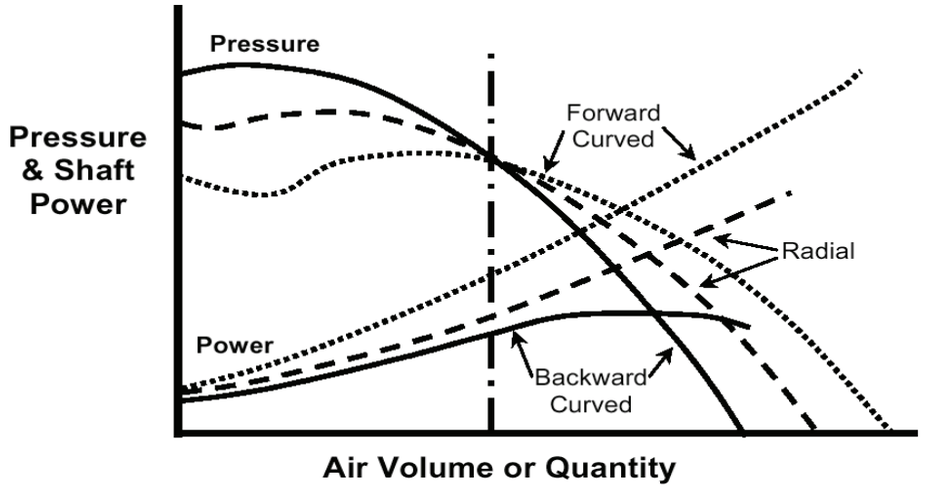


Figure 5.8 Fan Static Pressure and Power Requirements for Different Fans

Fan performance characteristics and efficiency differ based on fan and impeller type (See Figure 5.9).

In the case of centrifugal fans, the hub- to-tip ratios (ratio of inner-to-outer impeller diameter) the tip angles (angle at which forward or backward curved blades are curved at the blade tip - at the base the blades are always oriented in the direction of flow), and the blade width determine the pressure developed by the fan.

Forward curved fans have large hub-to-tip ratios compared to backward curved fans and produce lower pressure.

Radial fans can be made with different heel-to-tip ratios to produce different pressures.

At both design and off-design points, backward-curved fans provide the most stable operation. Also, the power required by most backward –curved fans will decrease at flow higher than design values. A similar effect can be obtained by using inlet guide vanes instead of replacing the impeller with different tip angles. Radial fans are simple in construction and are preferable for high-pressure applications.

Forward curved fans, however, are less efficient than backward curved fans and power rises continuously with flow. Thus, they are generally more expensive to operate despite their lower first cost.

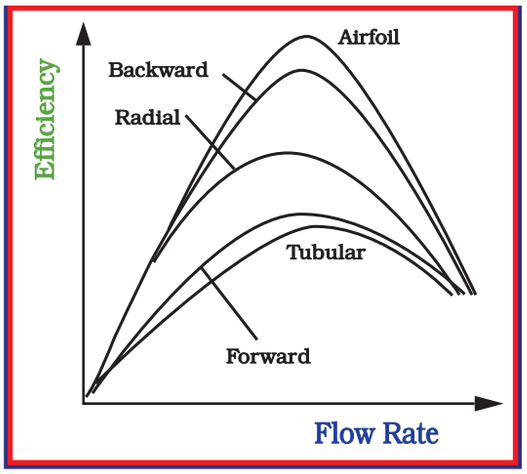


Figure 5.9 Fan Performance Characteristics based on Fans/ Impellers

Among centrifugal fan designs, aerofoil designs provide the highest efficiency (upto 10% higher than backward curved blades), but their use is limited to clean, dust-free air. Axial-flow fans produce lower pressure than centrifugal fans, and exhibit a dip in pressure before reaching the peak pressure point. Axial-flow fans equipped with adjustable / variable pitch blades are also available to meet varying flow requirements.

Propeller-type fans are capable of high-flow rates at low pressures. Tube-axial fans have medium pressure, high flow capability and are not equipped with guide vanes.

Vane-axial fans are equipped with inlet or outlet guide vanes, and are characterized by high pressure, medium flow-rate capabilities.

Performance is also dependant on the fan enclosure and duct design. Spiral housing designs with inducers, diffusers are more efficient as compared to square housings. Density of inlet air is another important consideration, since it affects both volume flow-rate and capacity of the fan to develop pressure. Inlet and outlet conditions (whirl and turbulence created by grills, dampers, etc.) can significantly alter fan performance curves from that provided by the manufacturer (which are developed under controlled conditions). Bends and elbows in the inlet or outlet ducting can change the velocity of air, thereby changing fan characteristics (the pressure drop in these elements is attributed to the system resistance). All these factors, termed as System Effect Factors, should, therefore, be carefully evaluated during fan selection since they would modify the fan performance curve.

Centrifugal fans are suitable for low to moderate flow at high pressures, while axial-flow fans are suitable for low to high flows at low pressures. Centrifugal fans are generally more expensive than axial fans. Fan prices vary widely based on the impeller type and the mounting (direct-or-belt-coupled, wall-or-duct-mounted). Among centrifugal fans, aerofoil and backward-curved blade designs tend to be somewhat more expensive than forward-curved blade designs and will typically provide more favourable economics on a lifecycle basis. Reliable cost comparisons are difficult since costs vary with a number of application-specific factors. A careful technical and economic evaluation of available options is important in identifying the fan that will minimize lifecycle costs in any specific application.

Safety margin

The choice of safety margin also affects the efficient operation of the fan. In all cases where the fan requirement is linked to the process/other equipment, the safety margin is to be decided, based on the discussions with the process equipment supplier. In general, the safety margin can be 5 % over the maximum requirement on flow rate.

In the case of boilers, the induced draft (ID) fan can be designed with a safety margin of 20 % on volume and 30 % on head. The forced draft (FD) fans and primary air (PA) fans do not require any safety margins. However, safety margins of 10 % on volume and 20 % on pressure are maintained for FD and PA fans.

Some pointers on fan specification

The right specification of the parameters of the fan at the initial stage, is pre-requisite for choosing the appropriate and energy efficient fan.

The user should specify following information to fan manufacturer to enable right selection:

Design operating point of the fan – volume and pressure

Normal operating point - volume and pressure

Maximum continuous rating

Low load operation - This is particularly essential for units, which in the initial few years may operate at lower capacities, with plans for upgradation at a later stage. The initial low load and the later higher load operational requirements need to be specified clearly, so that the manufacturer can supply a fan which can meet both the requirements, with different sizes of impeller.

Ambient temperature – The ambient temperatures, both the minimum and maximum, are to be specified to the supplier. This affects the choice of the material of construction of the impeller.

The maximum temperature of the gas at the fan during upset conditions should be specified to the supplier. This will enable choice of the right material of the required creep strength.

Density of gas at different temperatures at fan outlet

Composition of the gas – This is very important for choosing the material of construction of the fan.

Dust concentration and nature of dust – The dust concentration and the nature of dust (e.g. bagasse – soft dust, coal – hard dust) should be clearly specified.

The proposed control mechanisms that are going to be used for controlling the fan.

The operating frequency varies from plant-to-plant, depending on the source of power supply. Since this has a direct effect on the speed of the fan, the frequency prevailing or being maintained in the plant also needs to be specified to the supplier.

Altitude of the plant

The choice of speed of the fan can be best left to fan manufacturer. This will enable him to design the fan of the highest possible efficiency. However, if the plant has some preferred speeds on account of any operational need, the same can be communicated to the fan supplier.

System Resistance and Pressure Drop

The system resistance has a major role in determining the performance and efficiency of a fan. The system resistance also changes depending on the process. For example, the formation of the coatings / erosion of the lining in the ducts, changes the system resistance marginally. In some cases, the change of equipment (e.g. Replacement of Multi-cyclones with ESP / Installation of low pressure drop cyclones in cement industry) duct modifications, drastically shift the operating point, resulting in lower efficiency. In such cases, to maintain the efficiency as before, the fan has to be changed.

Hence, the system resistance has to be periodically checked, more so when modifications are introduced and action taken accordingly, for efficient operation of the fan.

System resistance in the application of centrifugal fan is the resistance offered by all the process equipment and duct line connected to the inlet of the fan as well as at the out let of the fan, usually stacks. In the Figure 5.10 draft required to overcome the resistance offered by the boiler to draw the flue gases -10 mmWc (- ve sign indicates the suction), the draft required to draw the flue gases through the economiser, Air heater and dust control system are added to estimate the total resistance at the inlet of the ID fan (i.e. -10 – 30 -40 – 150 = - 230 mmWc). However the ID fan requires a minimum of 10 mmWc pressure to push the gases to the bottom of the stack from where the gases will be taken care by natural draft into atmosphere. Hence the overall resistance (pressure drop) to be build up by the ID fan is the difference between the outlet pressure and the inlet pressure [i.e. 10 – (-230) = 240 mmWc].

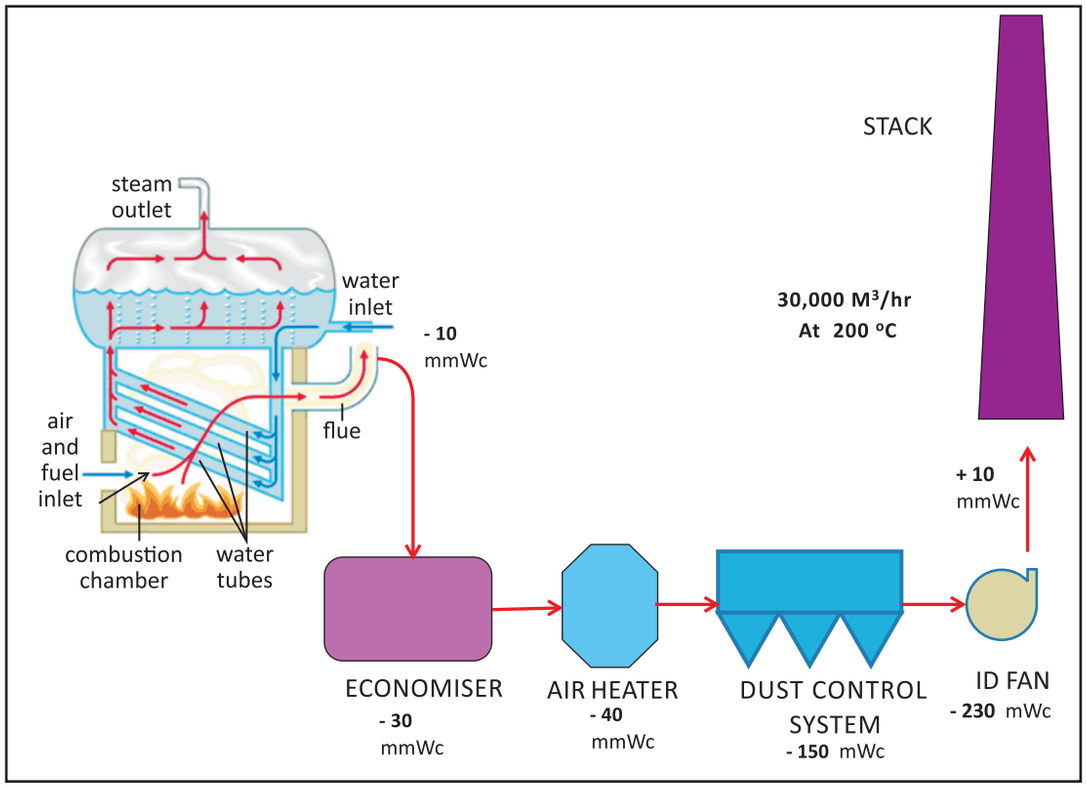


Figure 5.10 Pressure Drop across Various Equipment

System resistance is a function of gas density and the velocity of the gas. In the Figure 5.10 initially when there is no flue gas generation, there is no gas velocity in the process equipment and the ducts and hence the resistance (pressure drop) is zero (see Figure 5.11). After the boiler firing is started, there is continuous increase in the generation of flue gases and accordingly there is continuous increase in the velocity of the gases and continuous increase in draft. When the boiler is fired at rated capacity, the generation of flue gases are maximum resulting in corresponding pressure drop, which is the "operating point" (30,000 m^3/hr, 240 mmWc) of the boiler (Figure 5.11). Accordingly the ID fan with a characteristic (Performance) curve passing through the "operating point" of the system at the highest possible fan efficiency has to be selected for better utilisation of energy.

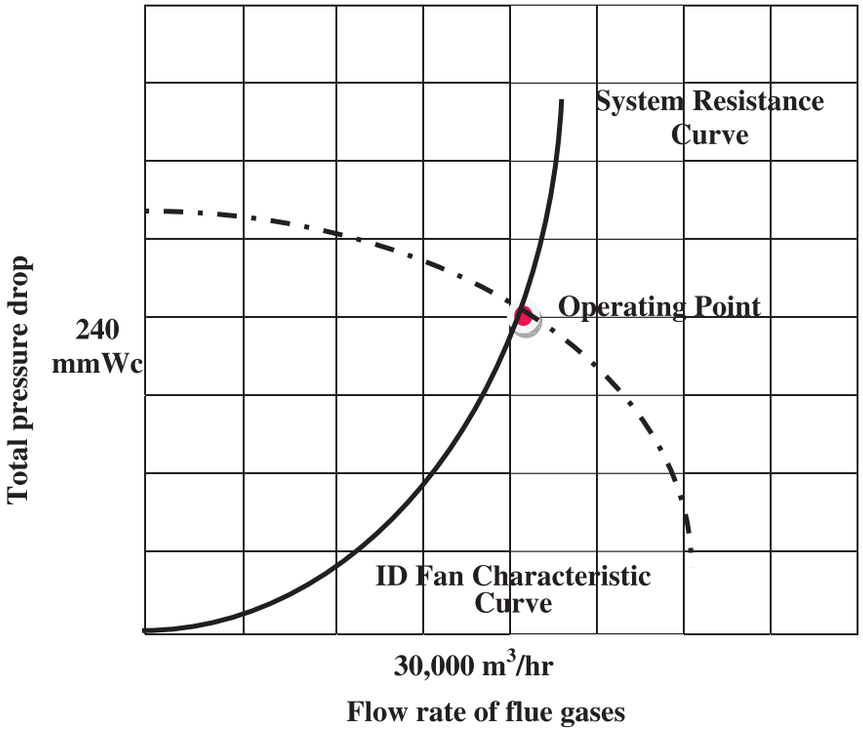


Figure 5.11 Fan System Curve

5.5 Flow Control Strategies

Typically, once a fan system is designed and installed, the fan operates at a constant speed. There may be occasions when a speed change is desirable, i.e., when adding a new run of duct that requires an increase in air flow (volume) through the fan. There are also instances when the fan is oversized and flow reductions are required.

Various ways to achieve change in flow are: pulley change, damper control, inlet guide vane control, variable speed drive and series and parallel operation of fans.

Pulley Change

When a fan volume change is required on a permanent basis, and the existing fan can handle the change in capacity, the volume change can be achieved with a speed change. The simplest way to change the

speed is with a pulley change. For this, the fan must be driven by a motor through a v-belt system. The fan speed can be increased or decreased with a change in the drive pulley or the driven pulley or in some cases, both pulleys. As shown in the Figure 5.12, a higher sized fan operating with damper control was downsized by reducing the motor (drive) pulley size from 8" to 6". The power reduction was 12 kW.

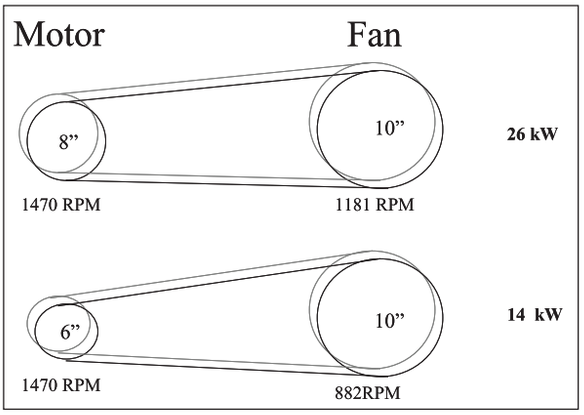


Figure 5.12 Pulley Change

Damper Controls

Some fans are designed with damper controls (see Figure 5.13). Dampers can be located at inlet or outlet. Dampers provide a means of changing air volume by adding or removing system resistance. This resistance forces the fan to move up or down along its characteristic curve, generating more or less air without changing fan speed. However, dampers provide a limited amount of adjustment, and they are not particularly energy efficient.

Figure 5.13 Damper change

Inlet Guide Vanes

Inlet guide vanes are another mechanism that can be used to meet variable air demand (see Figure 5.14). Guide vanes are curved sections that lay against the inlet of the fan when they are open. When they are closed, they extend out into the air stream. As they are closed, guide vanes pre-swirl the air entering the fan housing. This changes the angle at which the air is presented to the fan blades, which, in turn, changes the characteristics of the fan curve. Guide vanes are energy efficient for modest flow reductions – from 100 percent flow to about 80 percent. Below 80 percent flow, energy efficiency drops sharply.

Figure 5.14 Inlet Guide Vanes

Axial-flow fans can be equipped with variable pitch blades, which can be hydraulically or pneumatically controlled to change blade pitch, while the fan is at stationary. Variable-pitch blades modify the fan characteristics substantially and thereby provide dramatically higher energy efficiency than the other options discussed thus far.

Variable Speed Drives

Although, variable speed drives are expensive, they provide almost infinite variability in speed control. Variable speed operation involves reducing the speed of the fan to meet reduced flow requirements. Fan performance can be predicted at different speeds using the fan laws. Since power input to the fan changes as the cube of the flow, this will usually be the most efficient form of capacity control. However, variable speed control may not be economical for systems, which have infrequent flow variations. When considering variable speed drive, the efficiency of the control system (fluid coupling, eddy-current, VFD, etc.) should be accounted for, in the analysis of power consumption.

Series and Parallel Operation

Parallel operation of fans is another useful form of capacity control. Fans in parallel can be additionally equipped with dampers, variable inlet vanes, variable-pitch blades, or speed controls to provide a high degree of flexibility and reliability.

Combining fans in series or parallel can achieve the desired airflow without greatly increasing the system package size or fan diameter. Parallel operation is defined as having two or more fans blowing together side by side.

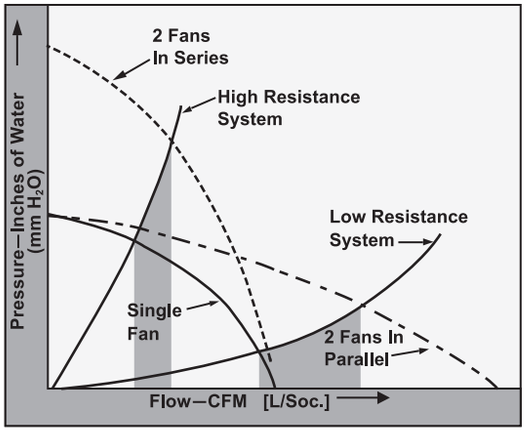


Figure 5.15 Series and Parallel Operation

The performance of two fans in parallel will result in doubling the volume flow, but only at free delivery. As Figure 5.15 shows, when a system curve is overlaid on the parallel performance curves, the higher the system resistance, the less increase in flow results with parallel fan operation. Thus, this type of application should only be used when the fans can operate in a low resistance almost in a free delivery condition.

Series operation can be defined as using multiple fans in a push-pull arrangement. By staging two fans in series, the static pressure capability at a given airflow can be increased, but again, not to double at every flow point, as the above Figure displays. In series operation, the best results are achieved in systems with high resistances.

In both series and parallel operation, particularly with multiple fans certain areas of the combined performance curve will be unstable and should be avoided. This instability is unpredictable and is a function of the fan and motor construction and the operating point.

Factors to be considered in the selection of flow control methods

Comparison on of various volume control methods with respect to power consumption (%) required power is shown in Figure 5.16.

All methods of capacity control mentioned above have turn-down ratios (ratio of maximum–to–minimum flow rate) determined by the amount of leakage (slip) through the control elements. For example, even with dampers fully closed, the flow may not be zero due to leakage through the damper. In the case of variable-speed drives the turn-down ratio is limited by the control system. In many cases, the minimum possible flow will be determined by the characteristics of the fan itself. Stable operation of a fan requires that it operate in a region where the system curve has a positive slope and the fan curve has a negative slope.

The range of operation and the time duration at each operating point also serves as a guide to selection of the most suitable capacity control system. Outlet damper control due to its simplicity, ease of operation, and low investment cost, is the most prevalent form of capacity control. However, it is the most inefficient of all methods and is best suited for situations where only small, infrequent changes are required (for example, minor process variations due to seasonal changes. The economic advantage

of one method over the other is determined by the time duration over which the fan operates at different operating points. The frequency of flow change is another important determinant. For systems requiring frequent flow control, damper adjustment may not be convenient. Indeed, in many plants, dampers are not easily accessible and are left at some intermediate position to avoid frequent control.

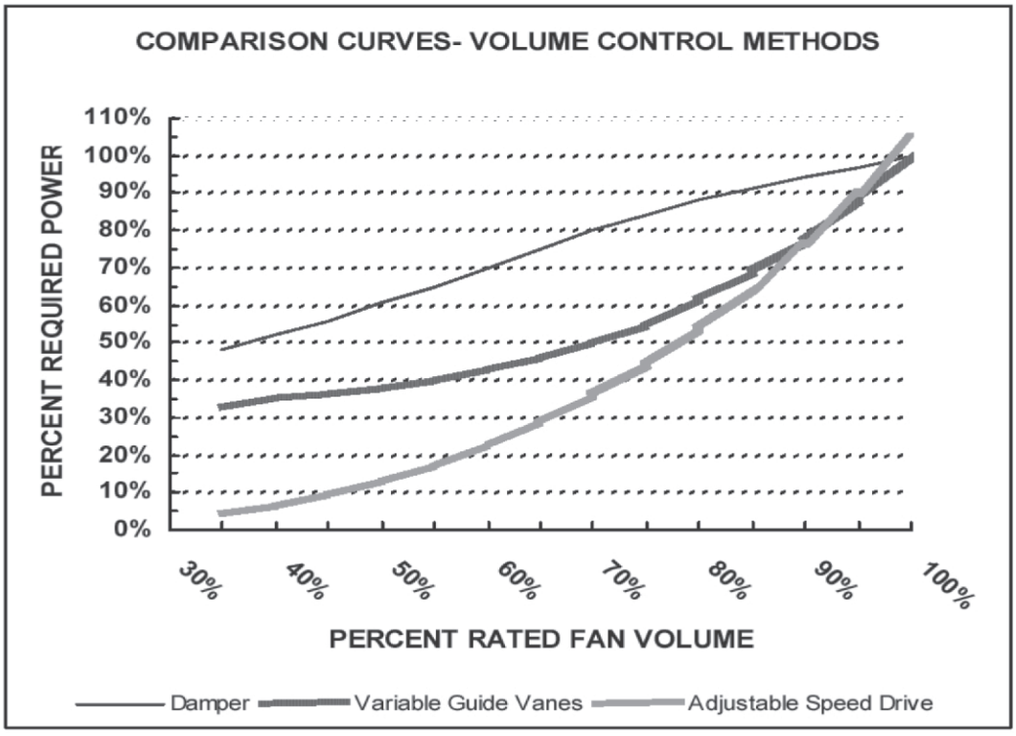


Figure 5.16 Comparison: Various Volume Control Methods

5.6 Fan Performance Assessment

The fans are tested for field performance by measurement of flow, head and temperature on the fan side and electrical motor kW input on the motor side.

Air flow measurement

Static pressure

Static pressure is the potential energy put into the system by the fan. It is given up to friction in the ducts and at the duct inlet as it is converted to velocity pressure. At the inlet to the duct, the static pressure produces an area of low pressure (see Figure 5.17).

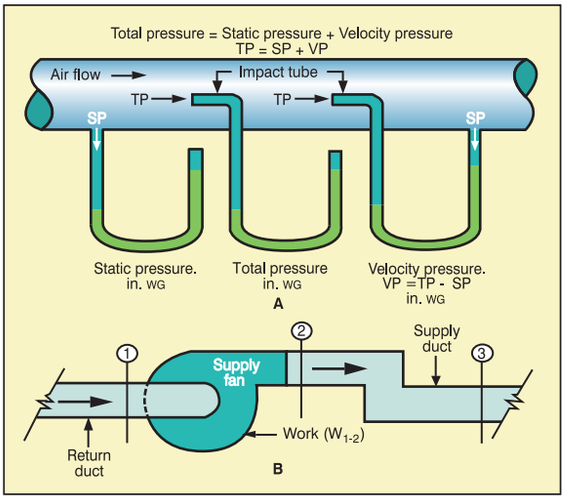


Figure 5.17 Static, Total and Velocity Pressure

Velocity pressure

Velocity pressure is the pressure along the line of the flow that results from the air flowing through the duct. The velocity pressure is used to calculate air velocity.

Total pressure

Total pressure is the sum of the static and velocity pressure. Velocity pressure and static pressure can change as the air flows though different size ducts accelerating and de-accelerating the velocity. The total pressure stays constant, changing only with friction losses. The illustration that follows shows how the total pressure changes in a system.

The fan flow is measured using pitot tube manometer combination, or a flow sensor (differential pressure instrument) or an accurate anemometer. Care needs to be taken regarding number of traverse points, straight length section (to avoid turbulent flow regimes of measurement) up stream and downstream of measurement location. The measurements can be on the suction or discharge side of the fan and preferably both where feasible.

Measurement by Pitot tube

The Figure 5.18 shows how velocity pressure is measured using a pitot tube and a manometer. Total pressure is measured using the inner tube of pitot tube and static pressure is measured using the outer tube of pitot tube. When the inner and outer tube ends are connected to a manometer, we get the velocity pressure. For measuring low velocities, it is preferable to use an inclined tube manometer instead of U tube manometer.

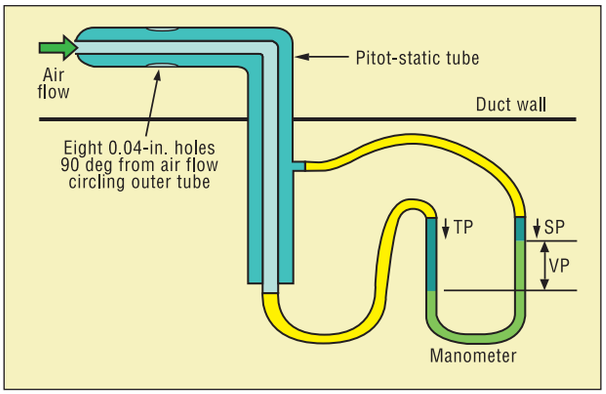


Figure 5.18 Velocity Measurement Using Pitot Tube

Measurements and Calculations

Velocity pressure/velocity calculation

When measuring velocity pressure the duct diameter (or the circumference from which to calculate the diameter) should be measured as well. This will allow us to calculate the velocity and the volume of air in the duct. In most cases, velocity must be measured at several places in the same system.

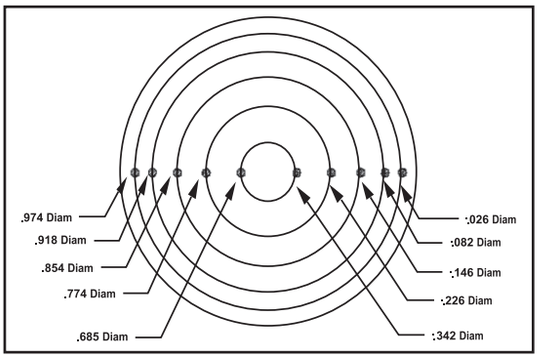


Figure 5.19 Traverse Points for Circular Duct

The velocity pressure varies across the duct. Friction slows the air near the duct walls, so the velocity is greater in the center of the duct. The velocity is affected by changes in the ducting configuration such as bends and curves. The best place to take measurements is in a section of duct that is straight for at least 3-5 diameters after any elbows, branch entries or duct size changes

To determine the average velocity, it is necessary to take a number of velocity pressure readings across the cross-section of the duct. The velocity should be calculated for each velocity pressure reading, and the average of the velocities should be used. Do not average the velocity pressure; average the velocities. For round ducts over 6 inches diameter, the following locations will give areas of equal concentric area (see Figure 5.19).

For best results, one set of readings should be taken in one direction and another set at a 90° angle to the first. For square ducts, the readings can be taken in 16 equally spaced areas. If it is impossible to traverse the duct, an approximate average velocity can be calculated by measuring the velocity pressure in the center of the duct and calculating the velocity. This value is reduced to an approximate average by multiplying by 0 .9.

Calculation of Velocity: After taking velocity pressures readings, at various traverse points, the velocity corresponding to each point is calculated using the following expression.

$$\text{Velocity (m/s)} = C_p \times \sqrt{\frac{2 \times 9.81 \times \Delta p}{\gamma}}$$

Where C_p = The pitot tube coefficient (Take manufacturer's value or assume 0.85)

$\triangle P$ = The average velocity pressure measured using pitot tube and inclined manometer by taking number of points over the entire cross-section of the duct, mm Water Column

γ = Gas density at flow conditions, kg/m^3

EXAMPLE

Air flow measurements using the pitot tube, in the primary air fan of a coal fired boiler gave the following data, calculate the velocity of air.

Air temperature = 38^{o}C
Velocity pressure = 47 mmWC
Pitot tube constant, Cp = 0.9
Air density at 38^{o}C = 1.135 kg /m^3

Find out the velocity of air in m/sec

$$\text{Velocity (m/s)} = C_p \times \sqrt{\frac{2 \times 9.81 \times \Delta p}{\gamma}}$$

$$= 0.9 \times \sqrt{\frac{2 \times 9.81 \times 47}{1.135}}$$

$$= 25.6 \text{ m/s}$$

Calculation of gas density

To calculate the velocity and volume from the velocity pressure measurements, it is necessary to know the density of gas. The density is dependent on altitude, temperature, molecular weight and pressure.

$$Density\,(\gamma), kg/m^3 = \frac{P \times M}{R \times T}$$

P – Absolute gas pressure, mmWC

M – Molecular weight of the gas, kg/kg mole (in the case of air M = 28.92 kg/kg mole)

T – Gas temperature, K

R – Gas constant, 847.84 mmWC m^3/kg mole K

Calculation of molecular weight for flue gas consisting of CO_2, CO, O_2, N_2 (M) (dry basis), kg/kg mole

$$= \{\%\ CO_2 \text{ x } M_{CO2} + \%\ O_2 \text{ x } M_{O2} + \%\ CO \text{ x } M_{CO} + \%\ N_2 \text{ x } M_{N2}\}/100$$

Volume calculation

The volume in a duct can be calculated for the velocity using the equation:

$$Volumetric\ flow\,(Q), m^3/\sec = Velocity, V\,(m/\sec)\ x\ Area\,(m^2)$$

Fan efficiency

Fan manufacturers generally use two ways to mention fan efficiency: mechanical efficiency (sometimes called the total efficiency) and static efficiency. Both measure how well the fan converts horsepower into flow and pressure.

The equation for determining mechanical efficiency is:

$$Fan\ \ Mechanical\ Efficiency\,(\eta_{mechanical}), \% = \frac{Volume\ in\ m^3/\sec \times \Delta p\,(total\ \ pressure)\ \ in\ mmWC}{102 \times Power\ input\ to\ \ fan\ shaft\ in\ \ kW}\ x\ 100$$

The static efficiency equation is the same except that the outlet velocity pressure is not added to the fan static pressure

$$Fan\ \ Static\ Efficiency\,(\eta_{static}), \% = \frac{Volume\ in\ m^3/\sec \times \Delta p\,(static\ \ pressure)\ \ in\ mmWC}{102 \times Power\ input\ to\ \ fan\ shaft\ in\ \ kW}\ x\ 100$$

Drive motor kW can be measured by a load analyzer. This kW multiplied by motor efficiency gives the shaft power to the fan.

5.7 Energy Savings Opportunities

Minimizing demands on the fan.

1. Minimising excess air level in combustion systems to reduce FD fan and ID fan load.
2. Minimising air in-leaks in hot flue gas path to reduce ID fan load, especially in case of

kilns, boiler plants, furnaces, etc. Cold air in-leaks increase ID fan load tremendously, due to density increase of flue gases and in-fact choke up the capacity of fan, resulting as a bottleneck for boiler / furnace itself.

3. In-leaks / out-leaks in air conditioning systems also have a major impact on energy efficiency and fan power consumption and need to be minimized.

The findings of performance assessment trials will automatically indicate potential areas for improvement, which could be one or a more of the following:

1. Change of impeller by a high efficiency impeller along with cone.
2. Change of fan assembly as a whole, by a higher efficiency fan
3. Impeller derating (by a smaller dia impeller)
4. Change of metallic / Glass reinforced Plastic (GRP) impeller by the more energy efficient hollow FRP impeller with aerofoil design, in case of axial flow fans, where significant savings have been reported
5. Fan speed reduction by pulley dia modifications for derating
6. Option of two speed motors or variable speed drives for variable duty conditions
7. Option of energy efficient flat belts, or, cogged raw edged V belts, in place of conventional V belt systems, for reducing transmission losses.
8. Adopting inlet guide vanes in place of discharge damper control
9. Minimizing system resistance and pressure drops by improvements in duct system

5.8 Case Study on Pressure Drop Reduction Across the Bag Filter

One of the Cement filter bag house is experiencing high Differential Pressure (DP) across the bag house while producing one particular type of cement. This high DP is resulting in high power consumption and puffing from the bag house. Upon examination it has been found that particle size distribution for this particular type of Cement associated with the fineness is creating high DP.

The results of this replacement are self explanatory in the details given below:

Application	Bag Filter for Cement Mill
Problem	High DP across the bag house while producing one particular type of Cement Product
Reason for the Problem	Characteristics of particle size distribution associated with its fineness is creating high DP across the bag house
Solution	Replace the existing filter bags with PTFE Membrane filter bags

Results	(a) Reduction of 50mm WC in DP across the bag house (b) Reduction of 5kWh in power consumption	
Parameter	Existing Bag	Bag with membrane
Gas Volume m3/hr	40000	41600
Temperature in ^{o} C (IL/OL)	96.8/86.9	98.2/88.7
Type of Filter bags	Mixed Felt	Mixed Felt with PTFE membrane
DP across the bag house mm WC	165	115
ID Fan power consumption kWh	51	46
Conclusion	The above results are categorically demonstrating that by changing the filter fabric we can achieve significant performance improvement of the bag house	

Case Studies VSD Applications

Case 1: Cement plants use a large number of high capacity fans. By using liners on the impellers, which can be replaced when they are eroded by the abrasive particles in the dust-laden air, the plants have been able to switch from radial blades to forward-curved and backward-curved centrifugal fans. This has vastly improved system efficiency without requiring frequent impeller changes.

For example, a careful study of the clinker cooler fans at a cement plant showed that the flow was much higher than required and also the old straight blade impeller resulted in low system efficiency. It was decided to replace the impeller with a backward-curved blade and use liners to prevent erosion of the blade. This simple measure resulted in a 53 % reduction in power consumption, which amounted to annual savings of Rs. 2.1 million.

Case 2: Another cement plant found that a large primary air fan which was belt driven through an arrangement of bearings was operating at system efficiency of 23 %. The fan was replaced with a direct coupled fan with a more efficient impeller. Power consumption reduced from 57 kW to 22 kW. Since cement plants use a large number of fans, it is generally possible to integrate the system such that air can be supplied from a common duct in many cases.

For example, a study indicated that one of the fans was operated with the damper open to only 5 %. By re-ducting to allow air to be supplied from another duct where flow was being throttled, it was possible to totally eliminate the use of a 55 kW fan.

Case 3: The use of variable-speed drives for capacity control can result in significant power savings. A 25 ton-per-hour capacity boiler was equipped with both an induced-draft and forced-draft fan. Outlet dampers were used to control the airflow. After a study of the air-flow pattern, it was decided to install a variable speed drive to control air flow. The average power consumption was reduced by nearly 41 kW resulting in annual savings of Rs. 0.33 million. The investment of Rs. 0.65 million for the variable-speed drive was paid back in under 2 years.

Case 4: The type of variable-speed drive employed also significantly impacts power consumption. Thermal power stations install a hydraulic coupling to control the capacity of the induced-draft fan. It was decided to install a VFD on ID fans in a 200 MW thermal power plant. A comparison of the power consumption of the two fan systems indicated that for similar operating conditions of flow and plant power generation, the unit equipped with the VFD control unit consumed, on average, 4 million units / annum less than the unit equipped with the hydraulic coupling.

5.9 Computational Fluid Dynamics

Use of Computational Fluid Dynamics (CFD) for Energy Efficiency

Computational Fluid Dynamics (CFD) is a powerful computer based simulation tool, which uses mathematics to model the physical system and to solve equations to predict the mass, momentum and energy. CFD tools helps to visualize the flow pattern, temperature and pressure profiles as well as particle movement inside the equipment / system under consideration. Being a simulation tool, various designs / operating parameters can be changed to arrive at best operating conditions for the plant. This also helps to determine the effect of changes in key process parameters before making any changes in the manufacturing process at plant. Since it is a proactive analysis and design tool, it can highlight the root cause not just the effect when evaluating plant problems. It is able to reduce scale-up problems because the models are based on fundamental physics and are scale independent. It is also useful in simulating conditions where it is not possible to take detailed measurements because of high temperature or dangerous environment.

Application of CFD in a Fan System for Energy Efficiency

In addition to optimization of the fan itself, the upstream and downstream flow situations are important for trouble free operation of the fan and maximum possible efficiency. If, for instance, the incoming flow is turbulent or swirling because of poorly designed bends or changes in cross-section, this will affect the fan's operating characteristics. Disturbances in the inflow and outflow zones have particularly serious effects in the case of high efficiency fans (> 80 %); as such fans depend on a non-swirling inflow in order to achieve their efficiency figures.

This case study describes the optimization of the induced draught flow of a double-inlet fan using CFD. Figure 5.20 shows the system arrangement drawing. The 3D model derived from this drawing is shown in Figure 5.21. The very sharp-edged transitions at inlet side affect the smooth flow.

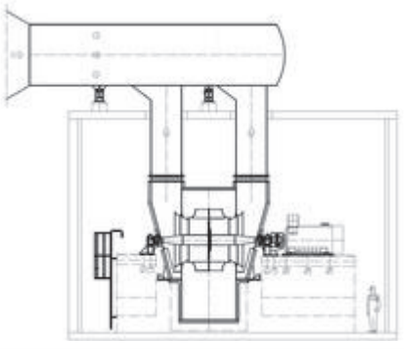


Figure 5.20 System arrangement drawing with induced draught fan

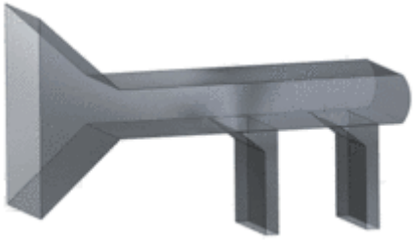


Figure 5.21 3D model of induced draught fan

The simulation (Figure 5.22) shows extensive flow disruption and instability. The pressure drop of this system is correspondingly high. From the inlet until the end of the first branch, the pressure drop is about 260 Pa. From the inlet until the next branch the drop is as high as 430 Pa because of the flow disruption near the end of the horizontal duct.

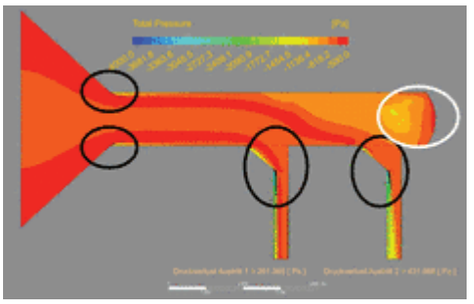


Figure 5.22 Total pressure profile with original design

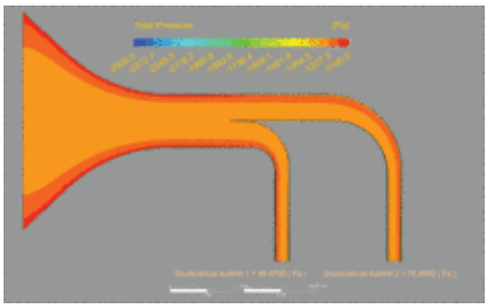


Figure 5.23 Total pressure profile after optimizing design

The black circles indicate unfavourable situations because this design involves transitions that are too sharp-edged and thus prevent the flow from following the contours, resulting in turbulence. This causes pressure drops and thus leads to additional electrical energy consumption. By contrast, the white circle indicates a duct end producing an inefficient fluid flow.

Since a portion of the vortex flow is sucked into the fan it has to cope with strongly swirling inflow air and therefore loses efficiency and suffers from mechanical vibrations.

Optimization of the geometry involved redesigning the critical points highlighted in Figure 5.22 so that the cross-section transitions were smooth. This redesigning succeeded in reducing the pressure drop at the front inflow duct to the fan by a factor of four, from 261 Pa to 66 Pa, while the pressure drop at the rear inflow duct that had been caused by the poorly-designed horizontal duct end (Figure 5.22, white circle) was even reduced to little more than one sixth the original figure. The severe swirling of the fan intake air (Figure 5.23, red circles) was also significantly reduced, enabling the fan to achieve its rated performance figures.

The mean pressure drop saving of 275 Pa at a volume flow of 5,00,000 m^3/h significantly reduces the power consumption. The resultant power saving was of the order of 49 kW.

Solved Example:

A V-belt centrifugal fan is supplying air to a process plant. The performance test on the fan gave the following parameters.

Density of air at 0 °C	1.293 kg/m^3
Ambient air temperature	40 °C
Diameter of the discharge air duct	0.8 m
Velocity pressure measured by Pitot tube in discharge duct	45 mmWC

Pitot tube coefficient	0.9
Static pressure at fan inlet	- 20 mmWC
Static pressure at fan outlet	185 mmWC
Power drawn by the motor coupled with the fan	75 kW
Belt transmission efficiency	97 %
Motor efficiency at the operating load	93 %

Find out the static fan efficiency.

Ans:

Air temperature	40 ^{o}C
Diameter of the discharge air duct	0.8 m
Velocity pressure measured by Pitot tube	45 mmWC
Static pressure at fan inlet	- 20 mmWC
Static pressure at fan outlet	185 mmWC
Power drawn by the motor	75 kW
Transmission efficiency	97%
Motor efficiency	93 %
Area of the discharge duct	3.14 x 0.8 x 0.8 x 1/4
	0.5024 m^2
Pitot tube coefficient	0.9
Corrected gas density	(273 x 1.293) / (273 + 40) = 1.1277
Volume	$\frac{C_p \times A\sqrt{2 \times 9.81 \times \triangle p \times \gamma}}{\gamma}$
	$\frac{0.9 \times 0.5024 \times \text{Sq rt.}(2 \times 9.81 \times 45 \times 1.1277)}{1.1277}$
	12.65 m^3/s
Power input to the shaft	75 x 0.97 x 0.93
	67.65 kW
Static Fan Efficiency % = $\frac{\text{Volume in } m^3 \text{ / Sec x total static pressure in mmwc}}{102 \times \text{Power input to the shaft in (kW)}}$	
Fan static Efficiency	$\frac{12.65 \times (185 - (-20)}{102 \times 67.65}$
	37.58 %

QUESTIONS	
Objective Type Questions	
1.	In a textile mill, two 150 cfm belt driven reciprocating compressors are seen to be working constantly with a loading time of 20 seconds and unloading time of 30 seconds. The best economic option for energy savings would be: a) switch off one compressor b) switch off one compressor and reduce motor pulley size of the other compressor appropriately c) adopt variable speed drive for one of the compressors d) none of the above
2.	The specific ratio as defined by ASME and used in differentiating fans, blowers and compressors, is given by a) discharge pressure/suction pressure b) suction pressure/discharge pressure c) discharge pressure/ (suction pressure + discharge pressure) d) suction pressure/ (suction pressure + discharge pressure)
3.	The efficiency of backward-inclined fans compared to forward curved fans is__ a) lower b) higher c) same d) none of the above
4.	For centrifugal fans, the relation between shaft input Power (kW) and Speed (N) is given by a) $\frac{kW_1}{kW_2} = \frac{N_1}{N_2}$ b) $\frac{kW_1}{kW_2} = \frac{N_1^2}{N_2^2}$ c) $\frac{kW_1}{kW_2} = \frac{N_1^3}{N_2^3}$ d) none of the above
5.	Parallel operation of two identical fans in a ducted system a) will double the flow b) will double the fan static pressure c) will increase flow by more than two times d) will not double the flow
6.	A fan is operating at 970 RPM developing a flow of 3000 $Nm^3/hr.$ at a static pressure of 650 mmWC. If the speed is reduced to 700 RPM, the static pressure (mmWC) developed will be a) 388.5 b) 244.3 c) 469 d) none of the above
7.	Identify the correct statement: a) the Specific Ratio of Compressors is higher than Blowers b) the Specific Ratio of Fans is higher than Blowers c) the Specific Ratio of Compressors is lower than Fans d) the Specific Ratio of Blowers is higher than Compressors
8.	Decreasing the rpm of a fan at partial loading by 10% results in: a) decrease of 10% in flow rate and decrease of 27% in power requirement b) decrease of 10% in flow rate and decrease of 19% in power requirement c) decrease of 10% in flow rate and increase of 10% in power requirement d) increase of 10% in flow rate and no appreciable change in power requirement
9.	The power drawn by a centrifugal fan is a) inversely proportional to fan efficiency b) directly proportional to fan efficiency c) inversely proportional to static pressure d) inversely proportional to flow rate

10.	The main reason for using Variable Frequency Drive (VFD) for capacity control in electrical motor driven centrifugal fans with fluctuating load is : a) improved power quality b) fan capacity is proportional to its speed whereas the power drawn by the fan is proportional to the cube of its speed c) improved power factor d) precise closed loop process control
	Short Type Questions
S-1	List any five energy conservation opportunities in a fan system.
S-2	An energy audit of a fan was carried out. It was observed that the fan was delivering 18,500 Nm^3/hr of air with static pressure rise of 45 mm WC. The power measurement of the 3-phase induction motor coupled with the fan recorded 2.9 kW/ phase on an average. The motor operating efficiency was assessed as 88% from the motor performance curves. What would be the fan static efficiency?
S-3	Explain the factors, which can change the system resistance?
S-4	A fan is operating at 900 RPM developing a flow of 3000 Nm^3/hr at a static pressure of 600 mmWC. What will be the flow and static pressure if the speed is reduced to 600 RPM?
S-5	Air flow measurements using the pitot tube, in the primary air fan of a coal fired boiler gave the following data Air temperature = 38°C Velocity pressure = 47 mmWC Pitot tube constant, Cp = 0.9 Air density at 0°C (standard data) = 1.293 kg /m^3 Find out the velocity of air in m/sec.
	Long Type Questions
L-1	A seal air fan for a coal mill is operating with suction damper in 25 % open condition. The power drawn at 50 Hz by fan motor is 120 kW. A VFD is to be installed eliminating the damper operation. It is found that the damper can be completely opened and the fan motor can be operated at 33 Hz. Calculated the power drawn by the fan motor at 33 Hz, assuming that motor and fan efficiency remains constant.
L-2	The size of an engine room to be ventilated is 30 m x 20 m x 5 m. The number of air changes per hour is designed to be 20. If the static pressure rise across the ventilator fan is 15 mm WC and fan efficiency is 70 % find out the motor power drawn at a motor efficiency of 90%.

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Chapter 6

Pumps and Pumping System

Learning Objectives

In this chapter you will learn about

- Types of pumps
- Significance of head flow curve and operating point of a pump
- Factors affecting pump performance
- Efficient pumping system operation
- Boiler feed water pump
- Municipal water, sewage and agricultural pumping systems
- Various flow control strategies
- Energy conservation opportunities in pumping systems

6. PUMPS AND PUMPING SYSTEM

Types, Performance evaluation, Efficient system operation, Flow control strategies, Energy conservation in boiler feed water pump, pumping systems for municipal drinking water, sewage water pumps, agriculture pump sets, energy conservation opportunities.

6.1 Pump Types

Pumps come in a variety of sizes for a wide range of applications. They can be classified according to their basic operating principle as dynamic or displacement pumps. Dynamic pumps can be sub-classified as centrifugal and special effect pumps. Displacement pumps can be sub-classified as rotary or reciprocating pumps.

In principle, any liquid can be handled by any of the pump designs. Where different pump designs could be used, the centrifugal pump is generally the most economical followed by rotary and reciprocating pumps. Although, positive displacement pumps are generally more efficient than centrifugal pumps, the benefit of higher efficiency tends to be offset by increased maintenance costs.

Since, worldwide, centrifugal pumps account for the majority of electricity used by pumps, the focus of this chapter is on centrifugal pump.

Centrifugal Pumps

A centrifugal pump (Figure 6.1) is of a very simple design. The two main parts of the pump are the impeller and the diffuser. Impeller, which is the only moving part, is attached to a shaft and driven by a motor. Impellers are generally made of bronze, polycarbonate, cast iron, stainless steel as well as other materials. The diffuser (also called as volute) houses the impeller and captures and directs the water off the impeller.

Water enters the center (eye) of the impeller and exits the impeller with the help of centrifugal force. As water leaves the eye of the impeller a low-pressure area is created, causing more water to flow into the eye. Atmospheric pressure and centrifugal force cause this to happen. Velocity is developed as the water flows through the impeller spinning at high speed. The water velocity is collected by the diffuser and converted to pressure by specially designed passageways that direct the flow to the discharge of the pump, or to the next impeller should the pump have a multi-stage configuration.

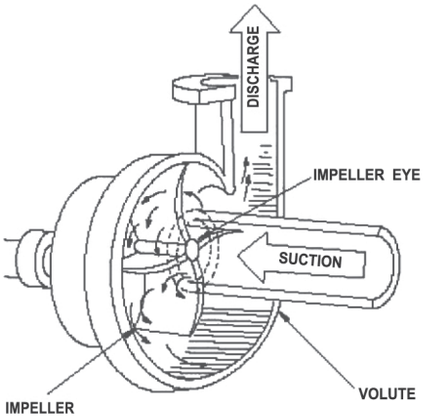


Figure 6.1 Centrifugal Pump

The pressure (head) that a pump will develop is in direct relationship to the impeller diameter, the number of impellers, the size of impeller eye, and shaft speed. Capacity is determined by the exit width of the impeller. The head and capacity are the main factors, which affect the horsepower size of the motor to be used. The more the quantity of water to be pumped, the more energy is required.

A centrifugal pump is not positive acting, it will not pump the same volume always. The greater the depth of the water, the lesser is the flow from the pump. Also, when it pumps against increasing pressure, the less it will pump. For these reasons it is important to select a centrifugal pump that is designed to do a particular job.

Since the pump is a dynamic device, it is convenient to consider the pressure in terms of head i.e. meters of liquid column. The pump generates the same head of liquid whatever the density of the liquid being pumped. The actual contours of the hydraulic passages of the impeller and the casing are extremely important, in order to attain the highest efficiency possible. The standard convention for centrifugal pump is to draw the pump performance curves showing Flow on the horizontal axis and Head generated on the vertical axis. Efficiency, Power & NPSH Required (described later), are also all conventionally shown on the vertical axis, plotted against Flow, as illustrated in Figure 6.2.

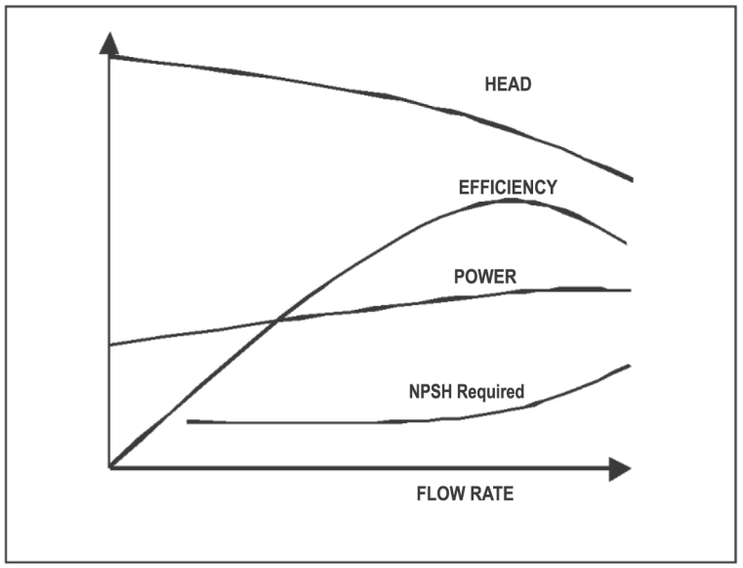


Figure 6.2 Pump Performance

Given the significant amount of electricity attributed to pumping systems, even small improvements in pumping efficiency could yield very significant savings of electricity. The pump is among the most inefficient of the components that comprise a pumping system, including the motor, transmission drive, piping and valves.

Hydraulic Power, Pump Shaft Power and Motor Input Power

Hydraulic Power P_h = Q (m^3/s) x Total Differential head, h_d - h_s (m) x ρ (kg/m^3) x g (m/s^2) / 1000

Where h_d - discharge head, h_s – suction head, ρ - density of the liquid, g – acceleration due to gravity

Pump Shaft Power P_s = Hydraulic power, P_h / Pump Efficiency, η_{Pump}

Motor Input Power = Pump shaft power P_s / Motor Efficiency, η_{Motor}

6.2 System Characteristics

In a pumping system, the objective, in most cases, is either to transfer a liquid from a source to a required destination, e.g. filling a high level reservoir or to circulate liquid around a system e.g. as a means of heat transfer in heat exchanger.

A pressure is needed to make the liquid flow at the required rate and this must overcome head 'losses' in the system. Losses are of two types: static and friction head.

Static head is simply the difference in height of the supply and destination reservoirs, as in Figure 6.3. In this illustration, flow velocity in the pipe is assumed to be very small. Another example of a system with only static head is pumping into a pressurized vessel with short pipe runs. Static head is independent of flow and graphically would be shown as in Figure 6.4.

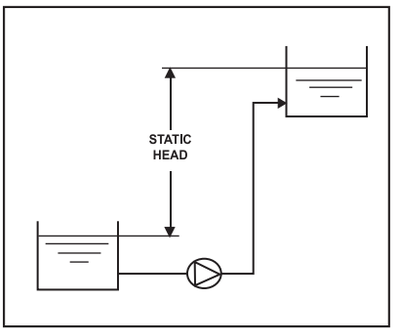


Figure 6.3 Static Head

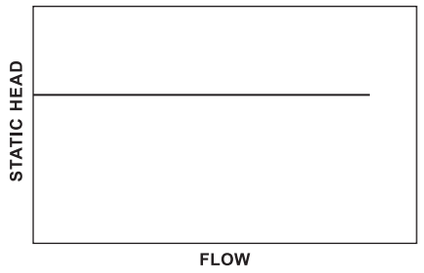


Figure 6.4 Static Head vs. Flow

Friction head (sometimes called dynamic head loss) is the friction loss, on the liquid being moved, in pipes, valves and equipment in the system. Friction tables are universally available for various pipe fittings and valves. These tables show friction loss per 100 feet (or meters) of a specific pipe size at various flow rates. In case of fittings, friction is stated as an equivalent length of pipe of the same size. The friction losses are proportional to the square of the flow rate. A closed loop circulating system without a surface open to atmospheric pressure, would exhibit only friction losses and would have a system friction head loss vs. flow curve as Figure 6.5.

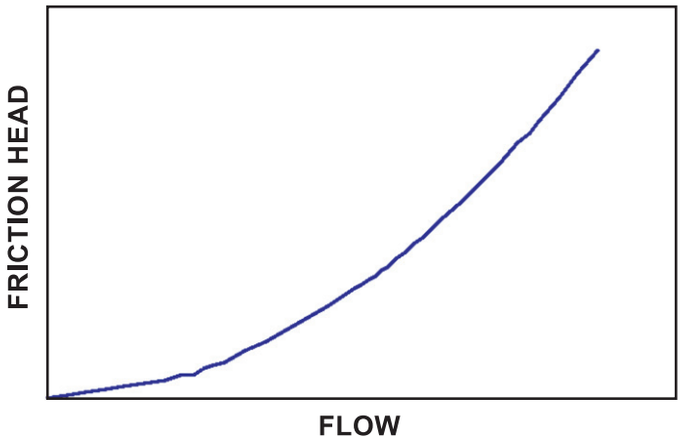


Figure 6.5 Friction Head vs. Flow

Most systems have a combination of static and friction head and the system curves for two cases are shown in Figures 6.6 and 6.7. The ratio of static to friction head over the operating range influences the benefits achievable from variable speed drives which shall be discussed later.

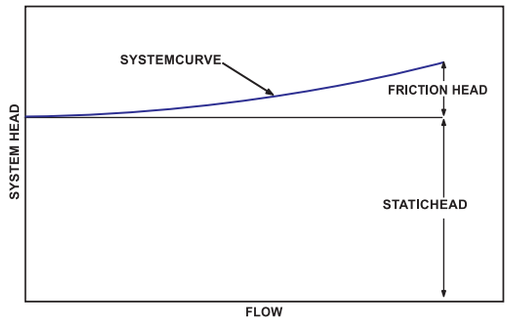


Figure 6.6 System with High Static Head

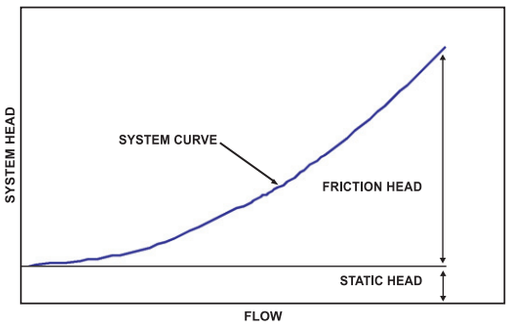


Figure 6.7 System with Low Static Head

Static head is a characteristic of the specific installation and reducing this head where this is possible generally helps both the cost of the installation and the cost of pumping the liquid. Friction head losses must be minimized to reduce pumping cost, but after eliminating unnecessary pipe fittings and length, further reduction in friction head will require larger diameter pipe, which adds to capital cost.

6.3 Pump Curves

The performance of a pump can be expressed graphically as head against flow rate. The centrifugal pump has a curve where the head falls gradually with increasing flow. This is called the pump characteristic curve (Head – Flow curve). See Figure 6.8.

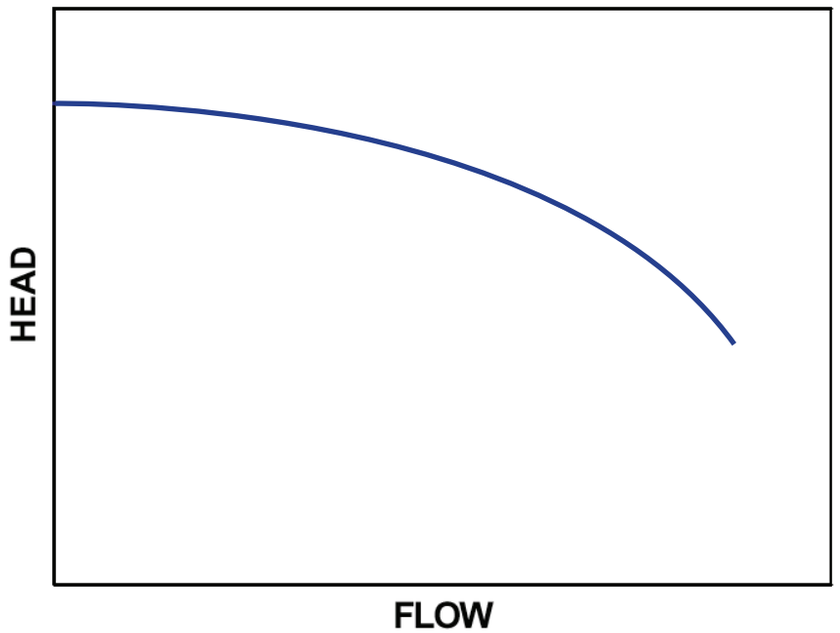


Figure 6.8 Head- Flow Curve

Pump operating point

When a pump is installed in a system the effect can be illustrated graphically by superimposing pump and system curves. The operating point will always be where the two curves intersect. Figure: 6.9.

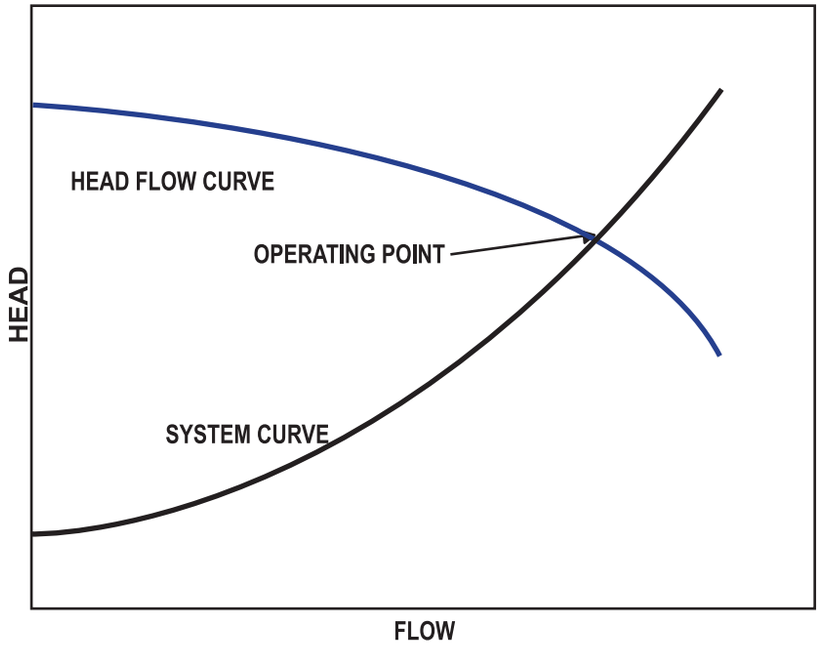


Figure 6.9 Pump Operating Point

If the actual system curve is different in reality to that calculated, the pump will operate at a flow and head different to that expected.

For a centrifugal pump, an increasing system resistance will reduce the flow, eventually to zero, but the maximum head is limited as shown. Even so, this condition is only acceptable for a short period without causing problems. An error in the system curve calculation is also likely to lead to a centrifugal pump selection, which is less than optimal for the actual system head losses. Adding safety margins to the calculated system curve to ensure that a sufficiently large pump is selected will generally result in installing an oversized pump, which will operate at an excessive flow rate or in a throttled condition, which increases energy usage and reduces pump life.

6.4 Factors Affecting Pump Performance

Matching Pump and System Head-flow Characteristics

Centrifugal pumps are characterized by the relationship between the flow rate (Q) they produce and the pressure (H) at which the flow is delivered. Pump efficiency varies with flow and pressure, and it is highest at one particular flow rate.

The Figure 6.10 below shows a typical vendor-supplied head-flow curve for a centrifugal pump. Pump head-flow curves are typically given for clear water. The choice of pump for a given application depends largely on how the pump head-flow characteristics match the requirement of the system downstream of the pump.

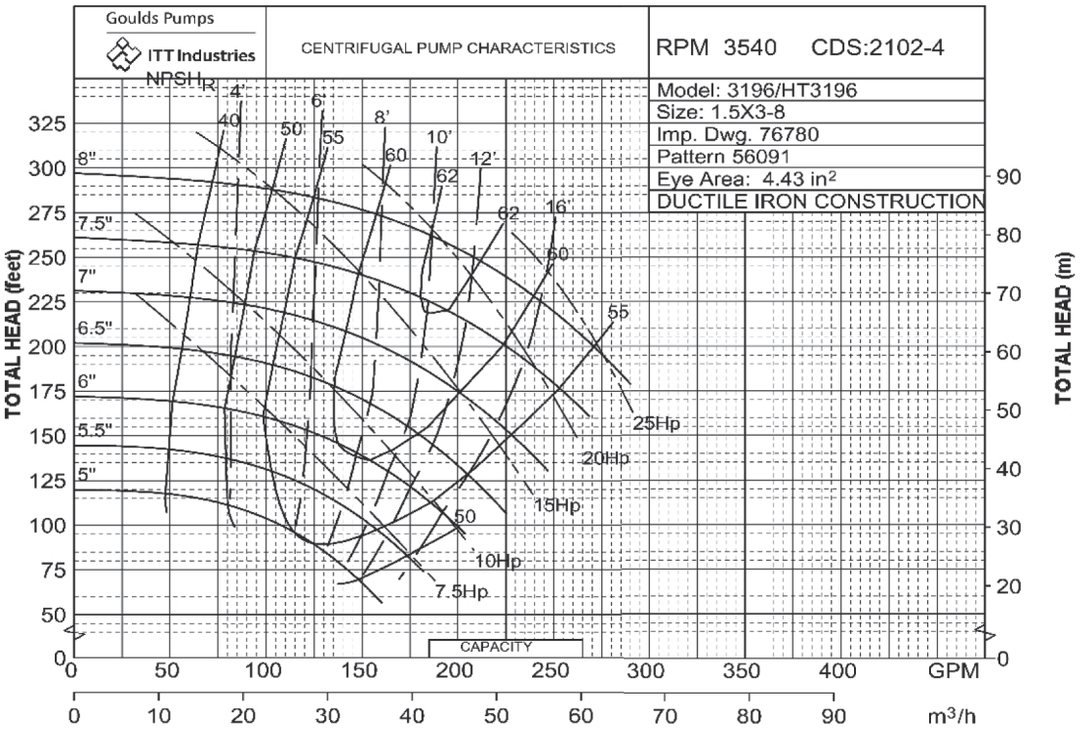


Figure 6.10 Typical Centrifugal Pump Performance Curve

Effect of over sizing the pump

As mentioned earlier, pressure losses to be overcome by the pumps are functions of flow – the system characteristics – are also quantified in the form of *head-flow curves*. The system curve is basically a plot of system resistance i.e. head to be overcome by the pump versus various flow rates. The system curves change with the physical configuration of the system; for example, the system curves depends upon height or elevation, diameter and length of piping, number and type of fittings and pressure drops across various equipment - say a heat exchanger.

A pump is selected based on how well the pump curve and system head-flow curves match. The pump operating point is identified as the point, where the system curve crosses the pump curve when they are superimposed on each other. The Figure 6.11 shows the effect on system curve with throttling.

In the system under consideration, water has to be first lifted to a height – this represents the static head.

Then, we make a system curve, considering the friction and pressure drops in the system- this is shown as the green curve.

Suppose, we have estimated our operating conditions as 500 m^3/hr flow and 50 m head, we will chose a pump curve which intersects the system curve (Point A) at the pump's *best efficiency point* (BEP).

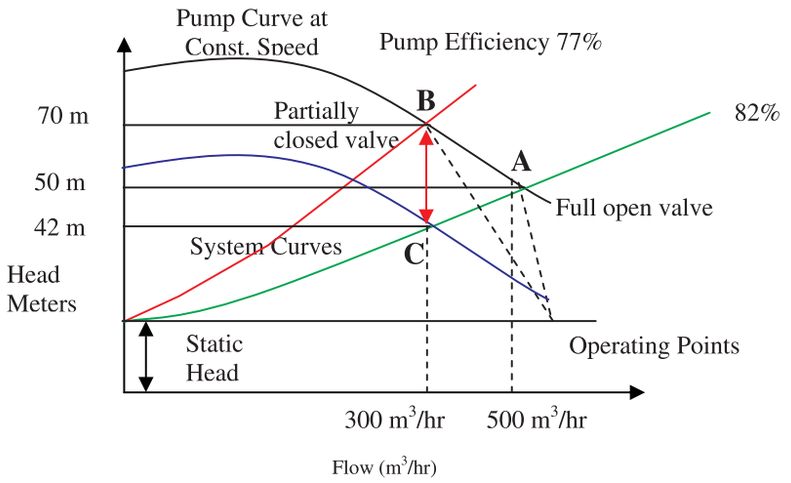


Figure 6.11 Effect on System Curve with Throttling

But, in actual operation, we find that 300 m^3/hr is sufficient. The reduction in flow rate has to be effected by a throttle valve. In other words, we are introducing an artificial resistance in the system.

Due to this additional resistance, the frictional part of the system curve increases and thus the new system curve will shift to the left - this is shown as the red curve.

So the pump has to overcome additional pressure in order to deliver the reduced flow. Now, the new system curve will intersect the pump curve at point B. The revised parameters are 300 m^3/hr at 70 m head. The red double arrow line shows the additional pressure drop due to throttling.

It may be noted that the best efficiency point has shifted from 82% to 77% efficiency. So it is actually needed to operate at point C, which is 300 m^3/hr on the original system curve. The head required at this point is only 42 meters.

Hence a new pump is needed, which will operate with its best efficiency point at C. But there are other simpler options rather than replacing the pump. The speed of the pump can be reduced or the existing impeller can be trimmed (or new lower size impeller). The blue pump curve represents either of these options.

Energy loss in throttling

Consider a case (see Figure 6.12) where we need to pump 68 m^3/hr of water at 47 m head. The pump characteristic curves (A…E) for a range of pumps are given in the Figure 6.12.

If we select pump E, then the efficiency is 60%

Hydraulic Power = Q (m^3/s) x Total Differential head, $h_d - h_s$ (m) x ρ (kg/m^3) x g (m/s^2) / 1000

$$= \frac{(68/3600) \times 47 \times 1000 \times 9.81}{1000}$$

= 8.7 kW

Shaft Power = 8.7 / 0.60 = 14.5 kW

Motor Power = 14.5 / 0.9 = 16.1 kW (considering a motor efficiency of 90%)

If we select pump A, then the efficiency is 50% (drop from earlier 60%)

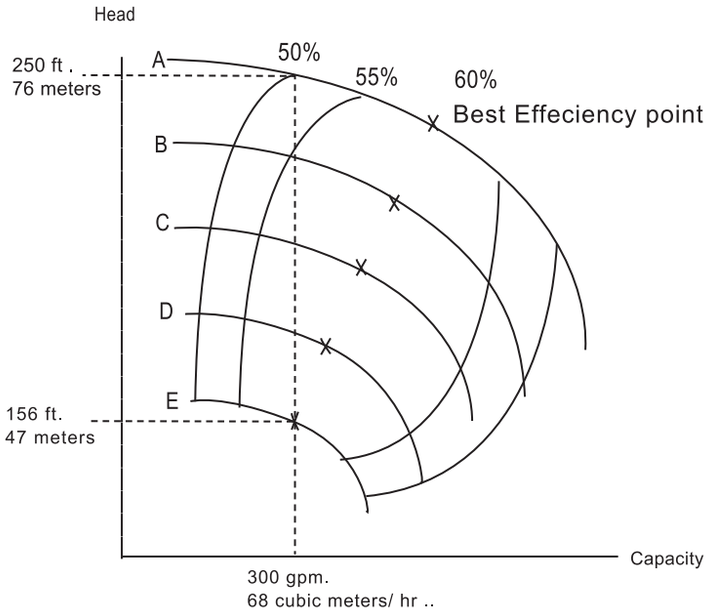


Figure 6.12 Pump Characteristic Curves

Obviously, this is an oversize pump. Hence, the pump has to be throttled to achieve the desired flow. Throttling increases the head to be overcome by the pump. In this case, head is 76 meters.

Hydraulic Power = Q (m^3/s) x Total Differential head, $h_d - h_s$ (m) x ρ (kg/m^3) x g (m/s^2) / 1000

$$= \frac{(68/3600) \times 76 \times 1000 \times 9.81}{1000}$$

= 14 kW

Shaft Power = 14 / 0.50 = 28 kW
Motor Power = 28 / 0.9 = 31 kW (considering a motor efficiency of 90%)

Hence, additional power drawn by A over E is 31 –16.1 = 14.9 kW.

Extra energy used = 8000 hrs/yr x 14.9 = 1,19,200 kWh/annum = Rs. 6,55,600/annum
(Rs. 5.50 per kWh)

In this example, the extra cost of the electricity is more than the cost of a new pump.

6.5 Efficient Pumping System Operation

To understand a pumping system, one must realize that all of its components are interdependent. When examining or designing a pump system, the process demands must first be established and most energy efficiency solution introduced. For example, does the flow rate have to be regulated continuously or in steps? Can on-off batch pumping be used? What is the flow rates needed and how are they distributed in time?

The first step to achieve energy efficiency in pumping system is to target the end-use. A plant water balance would establish usage pattern and highlight areas where water consumption can be reduced or optimized. Good water conservation measures, alone, may eliminate the need for some pumps.

Once flow requirements are optimized, then the pumping system can be analyzed for energy conservation opportunities. Basically this means matching the pump to requirements by adopting proper flow control strategies. Common symptoms that indicate opportunities for energy efficiency in pumps are given in the Table 6.1.

Table 6.1 Symptoms that Indicate Potential Opportunity for Energy Savings

Symptom	Likely Reason	Best Solutions
Throttle valve-controlled systems	Oversized pump	Trim impeller, smaller impeller, variable speed drive, two speed drive, lower RPM
Bypass line (partially or completely) open	Oversized pump	Trim impeller, smaller impeller, variable speed drive, two speed drive, lower RPM
Multiple parallel pump system with the same number of pumps always operating	Pump use not monitored or controlled	Install controls
Constant pump operation in a batch environment	Wrong system design	On-off controls
High maintenance cost (seals, bearings)	Pump operated far away from BEP	Match pump capacity with system requirement

Effect of speed variation

As stated above, a centrifugal pump is a dynamic device with the head generated from a rotating impeller. There is therefore a relationship between impeller peripheral velocity and generated head. Peripheral velocity is directly related to shaft rotational speed, for a fixed impeller diameter and so varying the rotational speed has a direct effect on the performance of the pump. All the parameters shown in figure 6.2 will change if the speed is varied and it is important to have an appreciation of how these parameters vary in order to safely control a pump at different speeds. The equations relating rotodynamic pump performance parameters of flow, head and power absorbed, to speed are known as the *Affinity Laws*:

$$Q \propto N$$

$$H \propto N^2$$

$$P \propto N^3$$

Where,

Q = Flow rate

H = Head

P = Power absorbed

N = Rotating speed

Efficiency is essentially independent of speed

Flow: Flow is proportional to the speed

$$Q_1 / Q_2 = N_1 / N_2$$

Example: $100 / Q_2 = 3000/1500$

$$Q_2 = 50 \text{ m}^3/\text{hr}$$

Head: Head is proportional to the square of speed

$$H_1/H_2 = N_1^2 / N_2^2$$

Example: $100 /H_2 = 3000^2 / 1500^2$

$$H_2 = 25 \text{ m}$$

Power (kW): Power is proportional to the cube of speed

$$kW_1 / kW_2 = N_1^3 / N_2^3$$

Example: $40/kW_2 = 3000^3 / 1500^3$

$$kW_2 = 5 \text{ kW}$$

As can be seen from the above laws, reduction in speed will result in considerable reduction in power consumption. This forms the basis for energy conservation in centrifugal pumps with varying flow requirements. The implication of this can be better understood as shown in an example of a centrifugal pump in Figure 6.13 below.

Points of equal efficiency on the curves for three different speeds are joined to make the iso-efficiency lines, showing that efficiency remains constant over small changes of speed providing the pump continues to operate at the same position related to its best efficiency point (BEP).

The affinity laws give a good approximation of how pump performance curves change with speed but in order to obtain the actual performance of the pump in a system, the system curve also has to be taken into account.

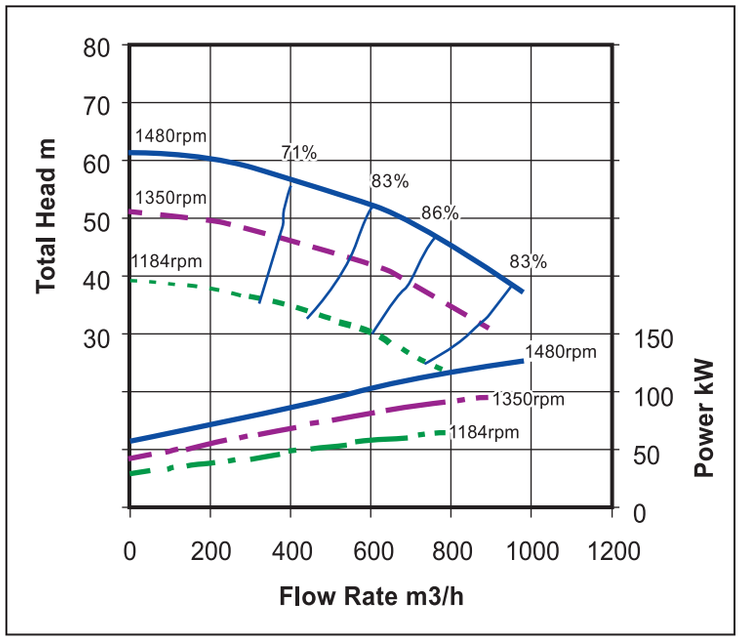


Figure 6.13 Example of Speed Variation Effecting Centrifugal Pump Performance

Effects of impeller diameter change

Changing the impeller diameter gives a proportional change in peripheral velocity, so it follows that there are equations, similar to the affinity laws, for the variation of performance with impeller diameter D:

$$Q \propto D$$

$$H \propto D^2$$

$$P \propto D^3$$

Efficiency varies when the diameter is changed within a particular casing. Note the difference in iso-efficiency lines in Figure 6.14 compared with Figure 6.13. The relationships shown here apply to the case for changing only the diameter of an impeller within a fixed casing geometry, which is a common practice for making small permanent adjustments to the performance of a centrifugal pump. Diameter changes are generally limited to reducing the diameter to about 75% of the maximum, i.e. a head reduction to about 50%. Beyond this, efficiency and NPSH are badly affected. However speed change can be used over a wider range without seriously reducing efficiency. For example reducing the speed by 50% typically results in a reduction of efficiency by 1 or 2 percentage points. The reason for the small loss of efficiency with the lower speed is that mechanical losses in seals and bearings, which generally represent <5% of total power, are proportional to speed, rather than speed cubed. It should be noted that if the change in diameter is more than about 5%, the accuracy of the squared and cubic relationships can fall off and for precise calculations, the pump manufacturer's performance curves should be referred to.

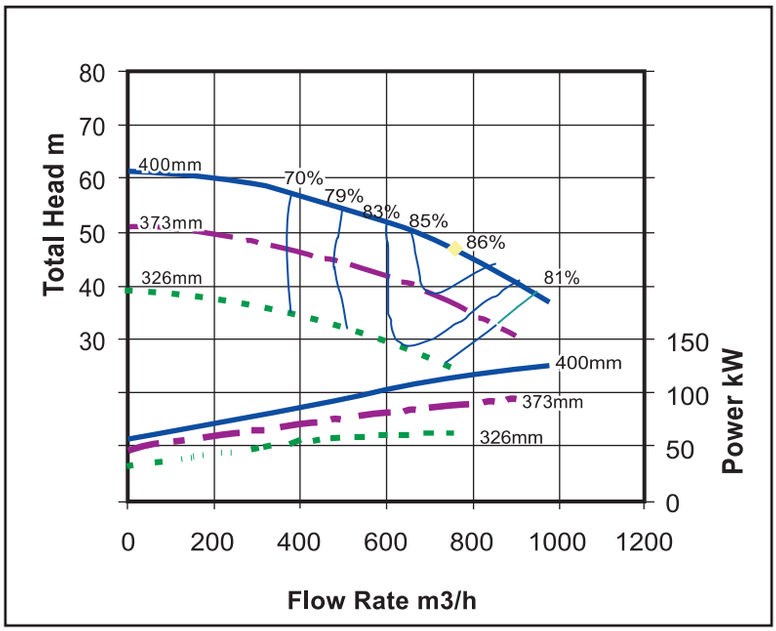


Figure 6.14 Example: Impeller Diameter Reduction on Centrifugal Pump Perfomance

The illustrated curves are typical of most centrifugal pump types. Certain high flow, low head pumps have performance curve shapes somewhat different and have a reduced operating region of flows. This requires additional care in matching the pump to the system when changing speed and diameter.

Pump suction performance

Liquid entering the impeller eye turns and is split into separate streams by the leading edges of the impeller vanes, an action which locally drops the pressure below that in the inlet pipe to the pump.

If the incoming liquid is at a pressure with insufficient margin above its vapour pressure then vapour cavities or bubbles appear along the impeller vanes just behind the inlet edges. This phenomenon is known as cavitation and has three undesirable effects:

1) The collapsing cavitation bubbles can erode the vane surface, especially when pumping water-based liquids.

2) Noise and vibration are increased, with possible shortened seal and bearing life.

3) The cavity areas will initially partially choke the impeller passages and reduce the pump performance. In extreme cases, total loss of pump developed head occurs.

The value, by which the liquid pressure at the eye of pump exceeds the liquid vapour pressure, is expressed as a head of liquid and referred to as Net Positive Suction Head Available – (NPSHA). This is a characteristic of the system design. The value of NPSH needed at the pump suction to prevent the pump from cavitation is known as NPSH Required – (NPSHR). This is a characteristic of the pump design.

The three undesirable effects of cavitation described above begin at different values of NPSHA and generally there will be cavitation erosion before there is a noticeable loss of pump head. However for

a consistent approach, manufacturers and industry standards, usually define the onset of cavitation as the value of NPSHR when there is a head drop of 3% compared with the head with cavitation free performance. At this point cavitation is present and prolonged operation at this point will usually lead to damage. It is usual therefore to apply a margin by which NPSHA should exceed NPSHR.

As would be expected, the NPSHR increases as the flow through the pump increases, see fig 6.2. In addition, as flow increases in the suction pipework, friction losses also increase, giving a lower NPSHA at the pump suction, both of which give a greater chance that cavitation will occur. NPSHR also varies approximately with the square of speed in the same way as pump head and conversion of NPSHR from one speed to another can be made using the following equations.

$$Q \propto N$$

$$NPSHR \propto N^2$$

It should be noted however that at very low speeds there is a minimum NPSHR plateau, NPSHR does not tend to zero at zero speed. It is therefore essential to carefully consider NPSH in variable speed pumping.

6.6 Flow Control Strategies

Pump control by varying speed

To understand how the speed variation changes the duty point, the pump and system curves are over-laid. Two systems are considered, one with only friction loss and another where static head is high in relation to friction head. It will be seen that the benefits are different.

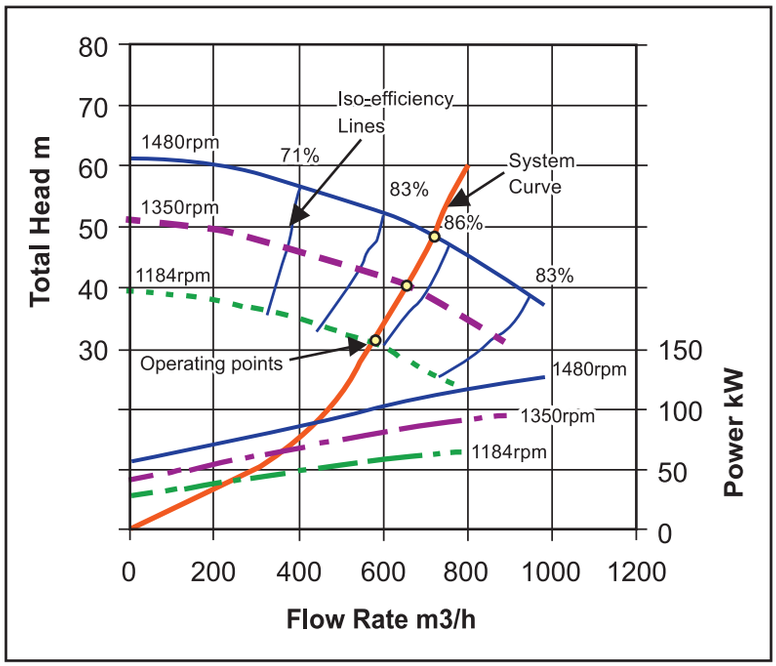


Figure 6.15 Example of the Effect of Pump Speed Change in a System with Only Friction Loss

In Figure 6.15, reducing speed in the friction loss system moves the intersection point on the system curve along a line of constant efficiency. The operating point of the pump, relative to its best efficiency point, remains constant and the pump continues to operate in its ideal region. The affinity laws are obeyed which means that there is a substantial reduction in power absorbed accompanying the reduction in flow and head, making variable speed the ideal control method for systems with friction loss.

In a system where static head is high, as illustrated in Figure 6.16, the operating point for the pump moves relative to the lines of constant pump efficiency when the speed is changed. The reduction in flow is no longer proportional to speed. A small turn down in speed could give a big reduction in flow rate and pump efficiency, which could result in the pump operating in a region where it could be damaged if it ran for an extended period of time even at the lower speed. At the lowest speed illustrated, (1184 rpm), the pump does not generate sufficient head to pump any liquid into the system, i.e. pump efficiency and flow rate are zero and with energy still being input to the liquid, the pump becomes a water heater and damaging temperatures can quickly be reached.

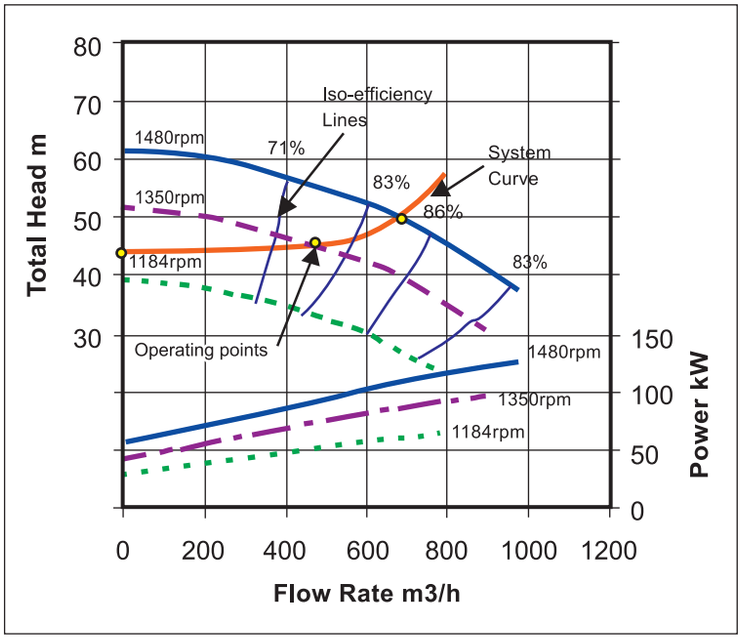


Figure 6.16 Example for the Effect of Pump Speed Change with a System with High Static Head

The drop in pump efficiency during speed reduction in a system with static head, reduces the economic benefits of variable speed control. There may still be overall benefits but economics should be examined on a case-by-case basis. Usually it is advantageous to select the pump such that the system curve intersects the full speed pump curve to the right of best efficiency, in order that the efficiency will first increase as the speed is reduced and then decrease. This can extend the useful range of variable speed operation in a system with static head. The pump manufacturer should be consulted on the safe operating range of the pump.

It is relevant to note that flow control by speed regulation is always more efficient than by control valve. In addition to energy savings there could be other benefits of lower speed. The hydraulic forces on the impeller, created by the pressure profile inside the pump casing, reduce approximately with the square of speed. These forces are carried by the pump bearings and so reducing speed increases bearing life. It can be shown that for a centrifugal pump, bearing life is inversely proportional to the 7^{th} power of speed. In addition, vibration and noise are reduced and seal life is increased providing the duty point remains within the allowable operating range.

The corollary to this is that small increases in the speed of a pump significantly increase power absorbed, shaft stress and bearing loads. It should be remembered that the pump and motor must be sized for the maximum speed at which the pump set will operate. At higher speed the noise and vibration from both pump and motor will increase, although for small increases the change will be small. If the liquid contains abrasive particles, increasing speed will give a corresponding increase in surface wear in the pump and pipework.

The effect on the mechanical seal of the change in seal chamber pressure should be reviewed with the pump or seal manufacturer, if the speed increase is large. Conventional mechanical seals operate satisfactorily at very low speeds and generally there is no requirement for a minimum speed to be specified, however due to their method of operation, gas seals require a minimum peripheral speed of 5 m/s.

Pumps in parallel switched to meet demand

Another energy efficient method of flow control, particularly for systems where static head is a high proportion of the total, is to install two or more pumps to operate in parallel. Variation of flow rate is achieved by switching on and off additional pumps to meet demand. The combined pump curve is obtained by adding the flow rates at a specific head. The head/flow rate curves for two and three pumps are shown in Figure 6.17.

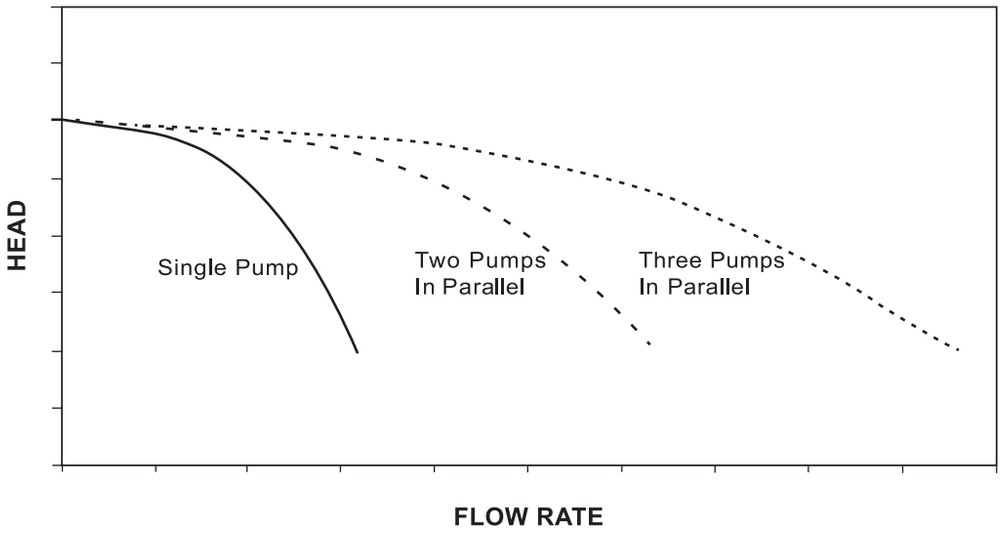


Figure 6.17 Typical Head-Flow Curves for Pumps in Parallel

The system curve is usually not affected by the number of pumps that are running. For a system with a combination of static and friction head loss, it can be seen, in Figure 6.18, that the operating point

of the pumps on their performance curves moves to a higher head and hence lower flow rate per pump, as more pumps are started. It is also apparent that the flow rate with two pumps running is not double that of a single pump. If the system head were only static, then flow rate would be proportional to the number of pumps operating.

It is possible to run pumps of different sizes in parallel provided their closed valve heads are similar. By arranging different combinations of pumps running together, a larger number of different flow rates can be provided into the system.

Care must be taken when running pumps in parallel to ensure that the operating point of the pump is controlled within the region deemed as acceptable by the manufacturer. It can be seen from Figure 6.18 that if 1 or 2 pumps were stopped then the remaining pump(s) would operate well out along the curve where NPSH is higher and vibration level increased, giving an increased risk of operating problems.

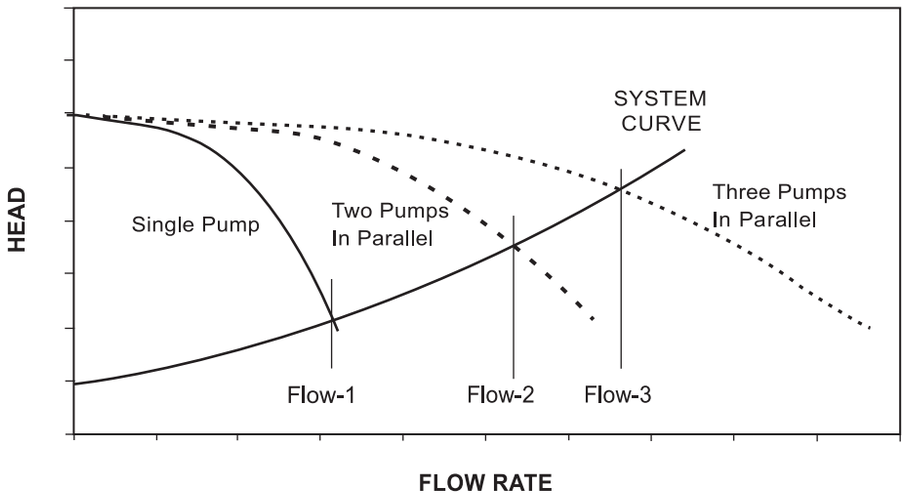


Figure 6.18 Typical Head-Flow Curves for Pumps in Parallel, With System Curve Illustrated.

Stop/start control

In this control method, the flow is controlled by switching pumps on or off. It is necessary to have a storage capacity in the system e.g. a reservoir, a wet well, an elevated tank or an accumulator type pressure vessel. The storage can provide a steady flow to the system with an intermittent operating pump. When the pump runs, it does so at the chosen (presumably optimum) duty point and when it is off, there is no energy consumption. If intermittent flow, stop/start operation and the storage facility are acceptable, this is an effective approach to minimize energy consumption.

The stop/start operation causes additional loads on the power transmission components and increased heating in the motor. The frequency of the stop/start cycle should be within the motor design criteria and checked with the pump manufacturer.

It may also be used to benefit from “off peak” energy tariffs by arranging the run times during the low tariff periods.

To minimize energy consumption with stop/start control, it is better to pump at as low flow rate as the process permits. This minimizes friction losses in the pipe and an appropriately small pump can be installed. For example, pumping at half the flow rate for twice as long can reduce energy consumption to a quarter. It means it is beneficial to run one pump at full capacity continuously rather than running two pumps at a time with a stop/start control.

Flow control valve

With this control method, the pump runs continuously and a valve in the pump discharge line is opened or closed to adjust the flow to the required value.

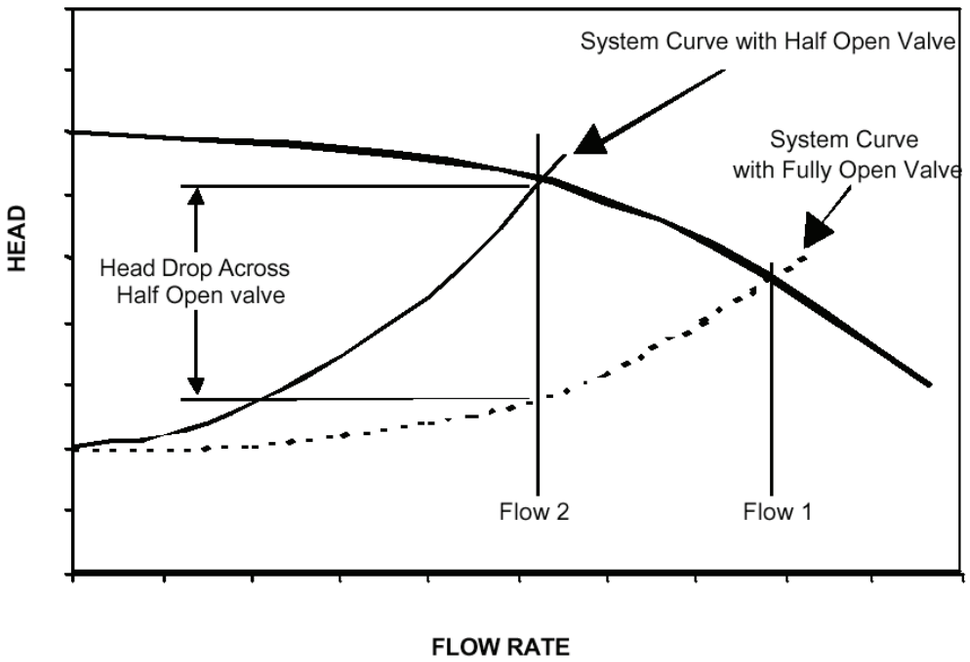


Figure 6.19 Control of Pump Flow by Changing System Resistance Using a Valve.

To understand how the flow rate is controlled, see Figure 6.19. With the valve fully open, the pump operates at "Flow 1". When the valve is partially closed, it introduces an additional friction loss in the system which is proportional to flow squared. The new system curve cuts the pump curve at "Flow 2" which is the new operating point. The head difference between the two curves is the pressure drop across the valve.

It is usual practice with valve control to have the valve 10% shut even at maximum flow. Energy is therefore wasted overcoming the resistance through the valve at all flow conditions. There is some reduction in pump power absorbed at the lower flow rate (see Figure 6.19), but the flow multiplied by the head drop across the valve, is wasted energy. It should also be noted that while the pump will accommodate changes in its operating point as far as it is able within its performance range, it can be forced to operate high on the curve where its efficiency is low and its reliability is affected.

Maintenance cost of control valves can be high, particularly on corrosive and solids-containing liquids. Therefore, the lifetime cost could be unnecessarily high.

By-pass control

With this control approach, the pump runs continuously at the maximum process demand duty with a permanent by-pass line attached to the outlet. When a lower flow is required the surplus liquid is bypassed and returned to the supply source.

An alternative configuration may have a tank supplying a varying process demand, which is kept full by a fixed duty pump running at the peak flow rate. Most of the time, the tank overflows and recycles back to the pump suction. This is even less energy efficient than a control valve because there is no reduction in power consumption with reduced process demand.

The small by-pass line sometimes installed to prevent a pump running at zero flow is not a means of flow control, but required for the safe operation of the pump.

Fixed Flow reduction

Impeller trimming

Impeller trimming refers to the process of machining the diameter of an impeller to reduce the energy added to the system liquid.

Impeller trimming offers a useful correction to pumps that, through overly conservative design practices or changes in system loads are oversized for their application.

Trimming an impeller provides a level of correction below buying a smaller impeller from the pump manufacturer. But in many cases, the next smaller size impeller is too small for the pump load. Also, smaller impellers may not be available for the pump size in question and impeller trimming is the only practical alternative short of replacing the entire pump/motor assembly. (See Figures 6.20 & 6.21 for before and after impeller trimming).

Impeller trimming reduces tip speed which in turn directly lowers the amount of energy imparted to the system liquid and lowers both the flow and pressure generated by the pump.

The Affinity Laws, which describe centrifugal pump performance, provide a theoretical relationship between impeller size and pump output (assuming constant pump speed):

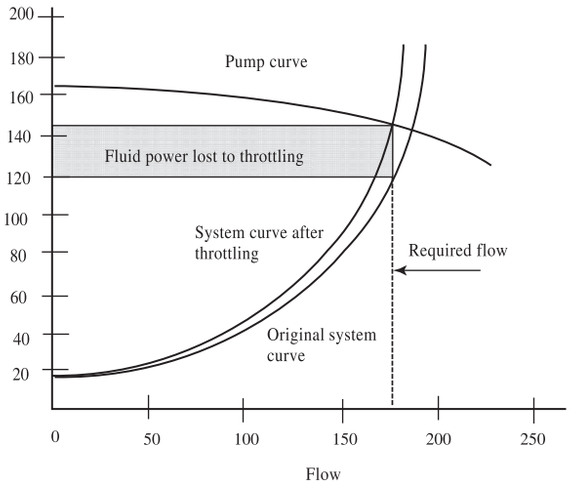


Figure 6.20 Before Impeller Trimming

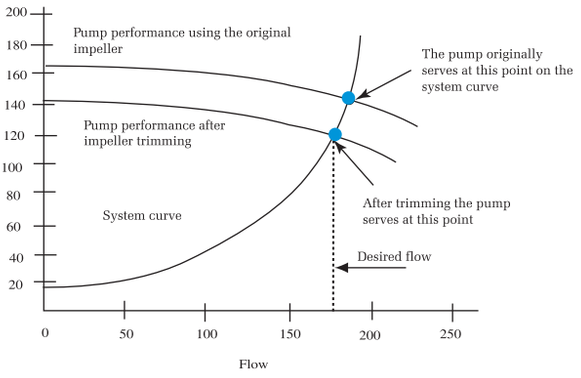


Figure 6.21 After Impeller Trimming

Where:

Q = flow

H = head

P = power

D = diameter of impeller

Subscript 1 = original pump,

Subscript 2 = pump after impeller trimming

$$Q_2 = \frac{D_2}{D_1} \times Q_1$$

$$H_2 = \left[\frac{D_2}{D_1}\right]^2 \times H_1$$

$$P_2 = \left[\frac{D_2}{D_1}\right]^3 \times P_1$$

Trimming an impeller changes its operating efficiency and the non-linearities of the Affinity Laws with respect to impeller machining complicate the prediction of pump performance. Consequently, impeller diameters are rarely reduced below 75 percent of their original size.

Meeting variable flow reduction

Variable Speed Drives (VSDs)

In contrast, pump speed adjustments provide the most efficient means of controlling pump flow. By reducing pump speed, less energy is imparted to the fluid and less energy needs to be throttled or bypassed. There are two primary methods of reducing pump speed: multiple-speed pump motors and variable speed drives (VSDs).

Although both directly control pump output, multiple-speed motors and VSDs serve entirely separate applications. Multiple-speed motors contain a different set of windings for each motor speed consequently they are more expensive and less efficient than single speed motors. Multiple speed motors also lack subtle speed changing capabilities within discrete speeds.

VSDs allow pump speed adjustments over a continuous range, avoiding the need to jump from speed to speed

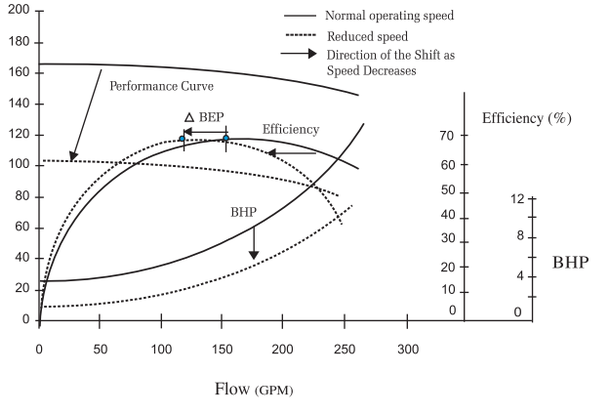


Figure 6.22 Effect of VFD

as with multiple-speed pumps. VSDs control pump speeds using several different types of mechanical and electrical systems. Mechanical VSDs include hydraulic clutches, fluid couplings and adjustable belts and pulleys. Electrical VSDs include eddy current clutches, wound-rotor motor controllers and variable frequency drives (VFDs). VFDs adjust the electrical frequency of the power supplied to a motor to change the motor's rotational speed. VFDs are by far the most popular type of VSD.

However, pump speed adjustment is not appropriate for all systems. In applications with high static head, slowing a pump risks inducing vibrations and creating performance problems that are similar to those found when a pump operates against its shutoff head. For systems in which the static head represents a large portion of the total head, caution should be used in deciding whether to use VFDs. Operators should review the performance of VFDs in similar applications and consult VFD manufacturers to avoid the damage that can result when a pump operates too slowly against high static head.

For many systems, VFDs offer a means to improve pump operating efficiency despite changes in operating conditions. The effect of slowing pump speed on pump operation is illustrated by the three curves in Figure 6.22. When a VFD slows a pump, its head/flow and power curves drop down and to the left and its efficiency curve shifts to the left. This efficiency response provides an essential cost advantage by keeping the operating efficiency as high as possible across variations in the system's flow demand, the energy and maintenance costs of the pump can be significantly reduced.

VFDs may offer operating cost reductions by allowing higher pump operating efficiency but the principal savings derive from the reduction in frictional or bypass flow losses. Using a system perspective to identify areas in which fluid energy is dissipated in non-useful work often reveals opportunities for operating cost reductions.

For example, in many systems, increasing flow through bypass lines does not noticeably impact the backpressure on a pump. Consequently, in these applications pump efficiency does not necessarily decline during periods of low flow demand. By analyzing the entire system, however, the energy lost in pushing liquid through bypass lines and across throttle valves can be identified.

Another system benefit of VFDs is a soft start capability. During startup, most motors experience in-rush currents that are 5 to 6 times higher than normal operating currents. This high current fades when the motor spins up to normal speed. VFDs allow the motor to be started with a lower startup current (usually only about 1.5 times the normal operating current). This reduces wear on the motor and its controller. VFDs will consume 4 to 6% power as a running cost apart from its initial cost.

6.7 Boiler Feed Water Pumps (BFP)

The feed water pumps are normally multi stage centrifugal pumps, sized based on boiler design pressure. The operation of a multistage pump is similar to the operation of several single stage pumps, of identical capacity, in series. Since most boilers operate below design pressure, the feed water pump head is often higher than required. This excessive pump head is dropped across pressure reducing valves and manual valves. Installing a VFD on the feed water pump in such cases can decrease pump power consumption and improve control performance. Trimming the impeller, reducing number of stages or changing the feed water pumps may also be feasible depending on variation in operating load of the boiler.

Boiler Feed Pump Control with VFD

There are several ways of controlling the pump

- **One pump, one boiler, no feed water regulating valve:** In this the pump speed is varied according to the level of water in the boiler. The level control system used for the feed water admission valve transmits its signal directly to the pump VFD controller. With this system it is possible to eliminate not only the feed pump constant discharge but also the boiler feed water regulating control valve and, thereby, cut initial capital investment. The inherent efficiency loss due to throttling is eliminated.
- **Constant discharge pressure control:** The feed pump is controlled to a predetermined pressure setting irrespective of plant load. The advantage of this system is that the pump will not be required to operate near shutoff pressures, due to the shifting of the operating point on the curve.
- **Constant differential pressure control:** Feed pump pressure is controlled to produce a predetermined pressure drop across the feed water regulating valve, usually approximately 3.5 to 5.5 kg/cm^2, thus allowing the boiler feed pump to follow plant demand.

Optimizing Boiler Feed Water Pump Capacity - Case

A waste heat boiler has two feed water pumps, each of 6 stages and having a capacity of 35 m^3/hr. The pumps are designed to generate a head of 276 m, normally one pump is operated

The actual steam demand is 28 TPH at 15 kg/cm^2. The capacity of the feed water pumps is far in excess of the requirement. This results in throttling of pump discharge leading to energy loss. To save energy in the BFW pumps, it was suggested to remove two impeller stages of the pump to effectively regulate the pressure developed in the pumps.

The impact of this measure on power consumption was then evaluated and the results are given in Table 6.2:

Table 6.2 Power Drawn Before and After

Operation of pumps	
With 6 stages (impellers)	53 kW
With 4 stages (impellers)	35 kW
Actual power savings achieved	18 kW

6.8 Municipal Water Pumping System

Municipal water pumps are predominantly centrifugal pumps and vertical turbine pumps. The capacity of a water pumping station is normally specified in Million Liters per Day (MLD) of water handled. Municipal water system consists of the following sub systems:

a) Raw water pump house, intake pumps at water source/river
b) Pure water pump house and filtration plant
c) Booster station as per the requirement
d) Elevated Storage Reservoirs in the distribution system

Vertical Turbine Pumps

Vertical turbine pump (Figure 6.23) [deep well turbine pump] is vertical axis centrifugal or mixed flow type pump comprising of stages which accommodate rotating impellers and stationary bowls possessing guide vanes.

These pumps are used where the pumping water level is below the limits of Volute centrifugal pump. They have higher initial cost and are more difficult to install and repair. The pressure head developed depends on the diameter of impeller and the speed at which it is rotated. The pressure head developed by single impeller is not great. Additional head is obtained by adding more bowl assemblies or stage.

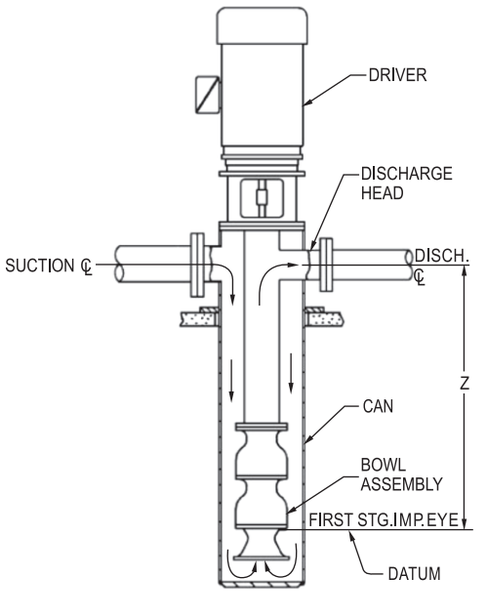


Figure 6.23 Vertical Turbine Pump

Construction:

It has three parts:

1. Pump Element:

The pump element is made up of one or more bowls or stages. Each bowl consists of an impeller and diffuser.

2. Discharge Column:

It connects the bowl assembly and pump head and conducts water from former to later.

Discharge head:

It consists of base from which the discharge column, bowl assembly and shaft assembly are suspended.

Submersible Pumps:

A vertical turbine pump close coupled to a small diameter submersible electric motor is termed as "submersible pump". The motor is fixed directly below the intake of the pump. The pump element and the motor operate under submerged condition. It can be used in very deep tube well where a long shaft would not be practical.

6.9 Sewage Water Pumps

The fundamental difference between a centrifugal sewage pump impeller and those of clear water pumps is its ability to pass solid material that would normally clog the latter. What differentiates various sewage pump impellers is the method by which they accomplish this. All sewage handling pumps comprise single suction impellers. This is to avoid the necessity of locating the pump shaft in the intake.

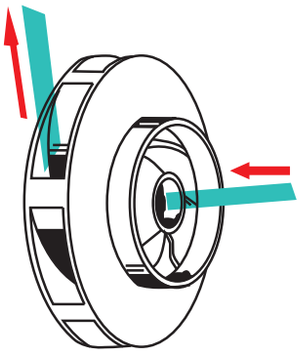


Figure 6.24 Radial Flow Impeller

The Figure 6.24 is that of a typical radial flow impeller of a pump used to handle coarse solids or fibrous matters. These pumps use volutes because diffuser is prone to clogging. The design of sewage pumps is largely determined by the size of foreign matter that must pass through the pump without clogging.

6.10 Agricultural Pumping System

The pumps used in agriculture sector are normally installed by individual farmers based on the guidelines provided by agriculture department/state utilities and feedback from other users. Most of the pumps used are locally manufactured keeping initial investment as the selection criteria rather than efficiency and energy conservation.

The pump sets used are generally inefficient with operating efficiency ranging from 30 - 55%. The wide variation is due to changing water levels in the intake thus forcing the pump to operate away from the best efficinecy point. The pump sets are more often oversized so as to draw water from increasingly declining depths and also to withstand large voltage fluctuations.

Mostly centrifugal pumps are used and the capacity of the pumps vary from 1 Hp to 25 HP . The rating of the pumps is decided based on water table levels. High rating pumps above 25 HP are also used in several areas. Large capacity cerifugal pumps of 75 HP to 500 HP ratings are also used by Irrgation departments to povide water to agricultural consumers. Diesel engine driven pumps are also common in areas where there is erratic or no power supply.

The following energy conservation opportunites have been demonstrated for energy savings in agricltural pumping.

- Installation of low friction foot valves
- Installation of low friction HDPE suction and delivery pipes
- Installation of long bends
- Installation of high efficency pumps and motors

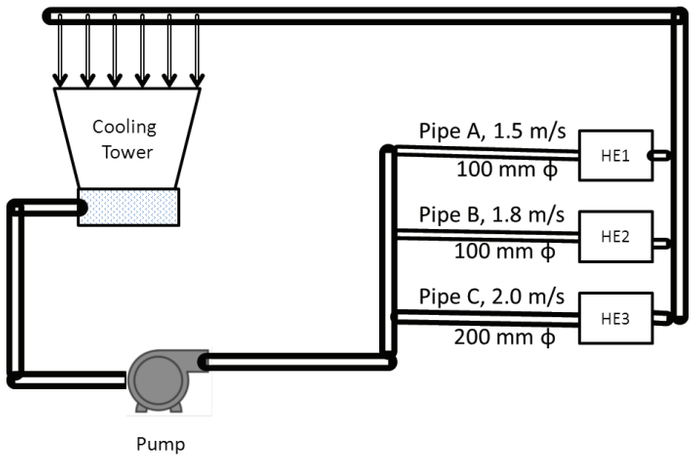
6.11 Energy Conservation Opportunities in Pumping Systems

- Ensure adequate NPSH at site of installation.
- Ensure availability of basic instruments at pumps like pressure gauges, flow meters.
- Operate pumps near Best Efficiency Point.
- Modify pumping system and pumps losses to minimize throttling.
- Adapt to wide load variation with variable speed drives or sequenced control of multiple units.
- Stop running multiple pumps - add an auto-start for an on-line spare or add a booster pump in the problem area.
- Use booster pumps for small loads requiring higher pressures.
- Increase liquid temperature differentials to reduce pumping rates in case of heat exchangers.
- Decrease outlet cold water temperature of cooling tower in order to reduce the pumping flow rates in case of mixing.

- Separate High Pressure and Low Pressure systems
- Repair seals and packing to minimize water loss by dripping.
- Balance the system to minimize flows and reduce pump power requirements.
- Avoid pumping head with a free-fall return (gravity); Use siphon effect to advantage:
- Conduct water balance to minimize water consumption.
- Avoid cooling water re-circulation in DG sets, air compressors, refrigeration systems, cooling towers feed water pumps, condenser pumps and process pumps.
- In multiple pump operations, carefully combine the operation of pumps to avoid throttling.
- Provide booster pump for few areas of higher head.
- Replace old pumps by energy efficient pumps.
- In the case of over designed pump, provide variable speed drive, or downsize / replace impeller or replace with correct sized pump for efficient operation.
- Optimize number of stages in multi-stage pump in case of head margins.
- Reduce system resistance by pressure drop assessment and pipe size optimization.

Solved Example:

The cooling water circuit of a process industry is depicted in the figure below. Cooling water is pumped to three heat exchangers via pipes A, B and C where flow is throttled depending upon the requirement. The diameter of pipes and measured velocities with non-contact ultrasonic flow meter in each pipe are indicated in the figure.



The following are the other data:

Measured motor power	:	50.7 kW
Motor efficiency at operating load	:	90%
Pump discharge pressure	:	3.4 kg/cm^2
Suction head	:	2 meters

Determine the efficiency of the pump.

Ans:

Flow in pipe A	22/7 x $(0.1)^2$/4 x 1.5
	0.011786 m^3/s
Flow in pipe B	22/7 x $(0.1)^2$/4 x 1.8
	0.014143 m^3/s
Flow in pipe C	22/7 x $(0.2)^2$/4 x 2.0
	0.062857 m^3/s
Total flow	0.088786 m^3/s
Total head	34 m – 2 m = 32 m
Pump hydraulic power	0.088786 x 32 x 9.81
	27.9 kW
	27.9 x 100/50.7 x 0.9
Pump efficiency	61 %

QUESTIONS	
Objective Type Questions	
1.	What is the impact on flow and pressure when the impeller of a pump is trimmed? a) flow decreases with increased pressure b) both flow and pressure increases c) both pressure and flow decreases d) none of the above
2.	The most efficient method of flow control in a pumping system is_____. a) throttling the flow b) speed control c) impeller trimming d) bypass control
3.	In case of centrifugal pumps, impeller diameter changes are generally limited to reducing the diameter to about _____ of maximum size. a) 75% b) 50% c) 25% d) none of the above
4.	Generally water pipe lines are designed with water velocity of a) < 1m/s b) up to 2 m/s c) > 2 m/s d) None of the above
5.	The head generated by a centrifugal pump is a) Independent of the density of the liquid being pumped. b) Directly proportional to the density of the liquid being pumped. c) Inversely proportional to the density of the liquid being pumped. d) Proportional to the square of the density of the liquid being pumped.
6.	Increasing the suction pipe diameter in a pumping system will a) reduce $NPSH_a$ b) Increase $NPSH_a$ c) decrease $NPSH_r$ d) Increase $NPSH_r$
7.	Small by-pass lines are installed some times to _____. a) control flow rate b) control pump delivery head c) prevent pump running at zero flow d) reduce pump power consumption
8.	Shaft power of the motor driving a pump is 30 kW. The motor efficiency is 0.9 and pump efficiency is 0.6. The power transmitted to the water is a) 16.2 kW b) 18.0 kW c) 27.0 kW d) none of the above
9.	In a centrifugal pump the velocity energy is converted to pressure by a) impeller b) casing c) throttle valve d) diffuser
10.	The friction loss in a pipe carrying a fluid is inversely proportional to the a) fluid flow b) square of the pipe diameter c) fluid velocity d) fifth power of pipe diameter
11.	It is possible to run pumps in parallel provided their_____ are similar a) suction head b) discharge heads c) closed valve heads d) none of the above

12.	If the speed of the pump is doubled, power goes up by a) 2 times b) 6 times c) 8 times d) 4 times
13.	The operating point in a pumping system is identified by a) point of intersection of system curve and efficiency curve b) point of intersection of pump curve and theoretical power curve c) point of intersection of pump curve and system curve d) none of the above
	Short Type Questions
S-1	What do mean by the term 'cavitation'? What are the undesirable effects of cavitation in a pumping system?
S-2	A pump is delivering 50 m^3/hr of water with a discharge pressure of 3.5 kg/cm^2. The water is drawn from a sump where water level is 5 meter below the pump centerline. The power drawn by the motor is 9.5 kW at 90% motor efficiency. Find out the pump efficiency.
S-3	A cooling water pump is connected to a pillar furnace. The specifications of the pump are as follows: Q = 12.5 lps H = 60 M p = 13.4 kW As per the pillar furnace manufacture, required quantity is 12.5 lps at 3.0 kg/cm^2. What type of energy conservation measure can be proposed and estimate the reduction in power consumption.
S-4	Briefly explain with a sketch the concept of pump head flow characteristics and system resistance.
S-5	What are the various methods of pump capacity control normally adopted?
	Long Type Questions
L-1	Explain parallel and series operation of pumps with necessary pump curves?
L-2	A centrifugal pump pumping water operates at 35 m^3/hr and at 1440 RPM. The pump operating efficiency is 68% and motor efficiency is 90%. The discharge pressure gauge shows 4.4 kg/cm^2. The suction is 2m below the pump centerline. If the speed of the pump is reduced by 50 % estimate the new flow, head and power.

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3. NPC energy audit reports
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Chapter 7

Cooling Tower

Learning Objectives

In this chapter you will learn about

- Cooling tower system
- Fanless cooling tower
- Performance parameters
  - Range
  - Approach
  - Cycles of Concentration
  - Effectiveness
- Cooling tower losses
- Factors affecting performance
- Cooling water treatment
- Energy saving opportunities

7. COOLING TOWER

Types, Fan less cooling tower, Performance evaluation, Assessment of cooling towers, Efficient system operation, Cooling water treatment, Flow control strategies and energy saving opportunities.

7.1 Introduction

Cooling towers are a very important part of many chemical plants. The primary task of a cooling tower is to reject heat into the atmosphere. They represent a relatively inexpensive and dependable means of removing low-grade heat from cooling water. The make-up water source is used to replenish water lost to evaporation. Hot water from heat exchangers is sent to the cooling tower. The water exits the cooling tower and is sent back to the exchangers or to other units for further cooling. Typical closed loop cooling tower system is shown in Figure 7.1.

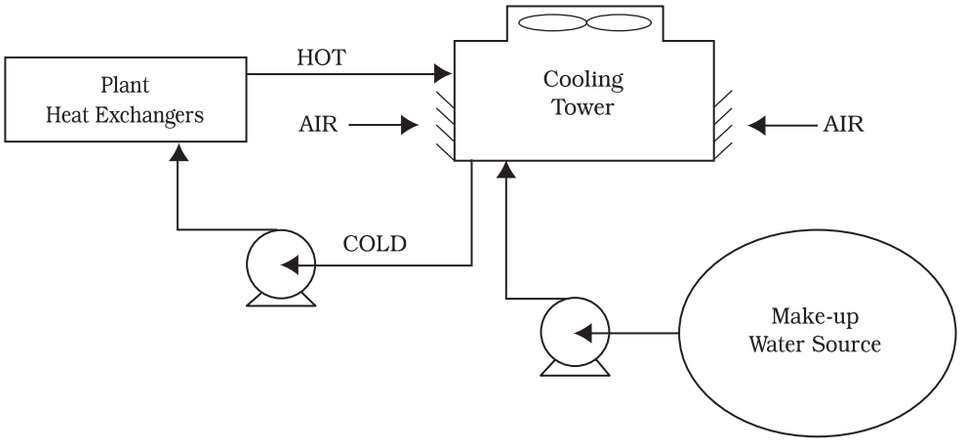


Figure 7.1 Cooling Water System

Cooling Tower Types

Cooling towers fall into two main categories: Natural draft and Mechanical draft.

Natural draft towers use very large concrete chimneys to introduce air through the media. Due to the large size of these towers, they are generally used for water flow rates above 45,000 m^3/hr. These types of towers are used only by utility power stations.

Mechanical draft towers utilize large fans to force or suck air through circulated water. The water falls downward over fill surfaces, which help increase the contact time between the water and the air - this helps maximise heat transfer between the two. *Cooling rates of Mechanical* draft towers depend upon their fan diameter and speed of operation. Since, the mechanical draft cooling towers are much more widely used; the focus is on them in this chapter.

Mechanical draft towers

Mechanical draft towers are available in the following airflow arrangements:

1. Counter flows induced draft.
2. Counter flow forced draft.
3. Cross flow induced draft.

In the counter flow induced draft design, hot water enters at the top, while the air is introduced at the bottom and exits at the top. Both forced and induced draft fans are used.

In cross flow induced draft towers, the water enters at the top and passes over the fill. The air, however, is introduced at the side either on one side (single-flow tower) or opposite sides (double-flow tower). An induced draft fan draws the air across the wetted fill and expels it through the top of the structure.

The Figure 7.2 illustrates various cooling tower types. Mechanical draft towers are available in a large range of capacities. Normal capacities range from approximately 10 tons, 2.5 m^3/hr flow to several thousand tons and m^3/hr. Towers can be factory built or field erected – for example concrete towers are only field erected.

Many towers are constructed so that they can be grouped together to achieve the desired capacity. Thus, many cooling towers are assemblies of two or more individual cooling towers or "cells." The number of cells they have, e.g., a eight-cell tower, often refers to such towers. Multiple-cell towers can be lineal, square, or round depending upon the shape of the individual cells and whether the air inlets are located on the sides or bottoms of the cells.

Components of Cooling Tower

The basic components of an evaporative tower are: Frame and casing, fill, cold water basin, drift eliminators, air inlet, louvers, nozzles and fans.

Frame and casing: Most towers have structural frames that support the exterior enclosures (casings), motors, fans, and other components. With some smaller designs, such as some glass fiber units, the casing may essentially be the frame.

Fill: Most towers employ fills (made of plastic or wood) to facilitate heat transfer by maximising water and air contact. Fill can either be splash or film type.

With splash fill, water falls over successive layers of horizontal splash bars, continuously breaking into smaller droplets, while also wetting the fill surface. Plastic splash fill promotes better heat transfer than the wood splash fill.

Film fill consists of thin, closely spaced plastic surfaces over which the water spreads, forming a thin film in contact with the air. These surfaces may be flat, corrugated, honeycombed, or other patterns. The film type of fill is the more efficient and provides same heat transfer in a smaller volume than the splash fill.

Cold water basin: The cold water basin, located at or near the bottom of the tower, receives the cooled water that flows down through the tower and fill. The basin usually has a sump or low point

for the cold water discharge connection. In many tower designs, the cold water basin is beneath the entire fill.

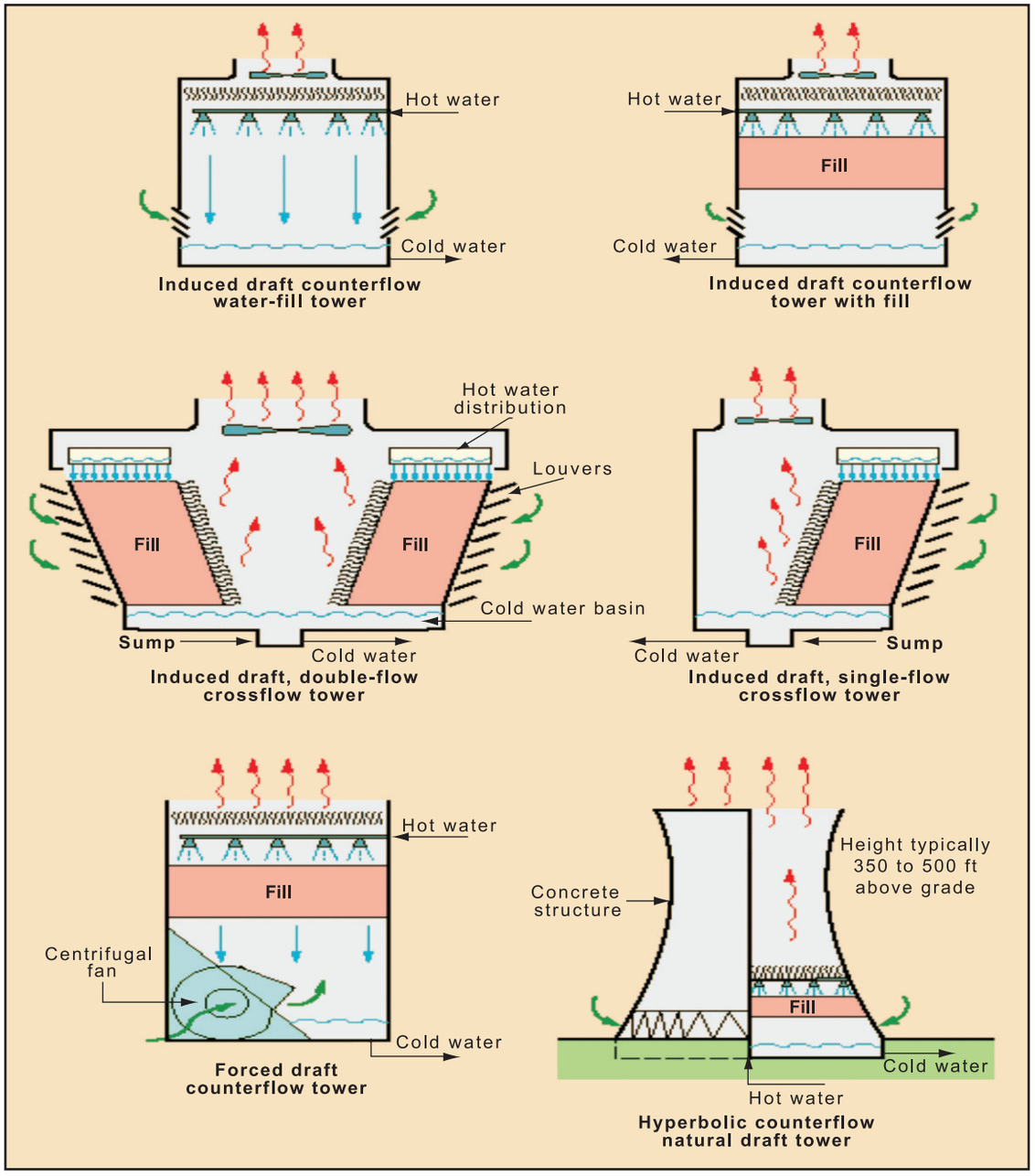


Figure 7.2 Cooling Tower Types

In some forced draft counter flow design, however, the water at the bottom of the fill is channeled to a perimeter trough that functions as the cold water basin. Propeller fans are mounted beneath the fill to blow the air up through the tower. With this design, the tower is mounted on legs, providing easy access to the fans and their motors.

Drift eliminators: These capture water droplets entrapped in the air stream that otherwise would be lost to the atmosphere.

Air inlet: This is the point of entry for the air entering a tower. The inlet may take up an entire side of a tower—cross flow design— or be located low on the side or the bottom of counter flow designs.

Louvers: Generally, cross-flow towers have inlet louvers. The purpose of louvers is to equalize air flow into the fill and retain the water within the tower. Many counter flow tower designs do not require louvers.

Nozzles: These provide the water sprays to wet the fill. Uniform water distribution at the top of the fill is essential to achieve proper wetting of the entire fill surface. Nozzles can either be fixed in place and have either round or square spray patterns or can be part of a rotating assembly as found in some circular cross-section towers.

Fans: Both axial (propeller type) and centrifugal fans are used in towers. Generally, propeller fans are used in induced draft towers and both propeller and centrifugal fans are found in forced draft towers. Depending upon their size, propeller fans can either be fixed or variable pitch.

A fan having non-automatic adjustable pitch blades permits the same fan to be used over a wide range of kW with the fan adjusted to deliver the desired air flow at the lowest power consumption.

Automatic variable pitch blades can vary air flow in response to changing load conditions.

Tower Materials

In the early days of cooling tower manufacture, towers were constructed primarily of wood. Wooden components included the frame, casing, louvers, fill, and often the cold water basin. If the basin was not of wood, it likely was of concrete.

Today, tower manufacturers fabricate towers and tower components from a variety of materials. Often several materials are used to enhance corrosion resistance, reduce maintenance, and promote reliability and long service life. Galvanized steel, various grades of stainless steel, glass fiber, and concrete are widely used in tower construction as well as aluminum and various types of plastics for some components.

Wood towers are still available, but they have glass fiber rather than wood panels (casing) over the wood framework. The inlet air louvers may be glass fiber, the fill may be plastic, and the cold water basin may be steel.

Larger towers sometimes are made of concrete. Many towers—casings and basins—are constructed of galvanized steel or, where a corrosive atmosphere is a problem, stainless steel. Sometimes a galvanized tower has a stainless steel basin. Glass fiber is also widely used for cooling tower casings and basins, giving long life and protection from the harmful effects of many chemicals.

Plastics are widely used for fill, including PVC, polypropylene, and other polymers. Treated wood splash fill is still specified for wood towers, but plastic splash fill is also widely used when water conditions mandate the use of splash fill. Film fill, because it offers greater heat transfer efficiency, is the fill of choice for applications where the circulating water is generally free of debris that could plug the fill passageways.

Plastics also find wide use as nozzle materials. Many nozzles are being made of PVC, ABS, polypropylene, and glass-filled nylon. Aluminum, glass fiber, and hot-dipped galvanized steel are commonly used fan materials. Centrifugal fans are often fabricated from galvanized steel. Propeller fans are fabricated from galvanized, aluminum, or molded glass fiber reinforced plastic.

Fanless Cooling Towers

Basis of Theory

Fanless cooling tower (Figure 7.3) takes advantage of the water pressure of the existing water circulation pump forming a water screen with specially designed ejection headers. As the water flows through the nozzles at high velocity, based on a ejector principle, low pressure is created which sucks the ambient cold air into the tower. The kinetic energy of Water entering the cooling tower is converted into kinetic energy of the air by the use of specially designed ejector nozzles. Water Pressure required in the Jet Ejector Nozzles is min. 0.5 Bar.

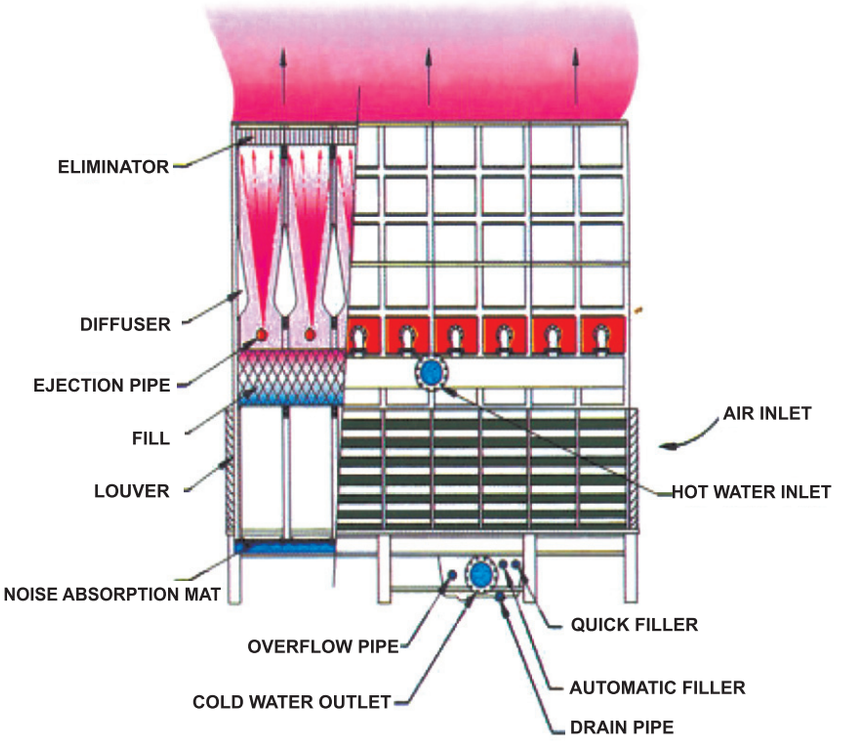


Figure 7.3 Fanless Cooling Tower

The incoming air passes through the fills at the bottom while the ejected water falls on the fills thus enabling a counter current heat exchange between water and air. Drift eliminators are provided to contain the drift losses.

Features of Fanless Cooling Tower

Energy saving

Since fans are not used in this type of cooling there is a considerable saving of power even though marginally higher power consumption is required for the pump.

Low noise

The noises of traditional cooling tower originate from the operating fans and motors. Further the vibration caused by these transmission units reinforces the noise resonance. This problem is eliminated in fanless cooling tower since no fan/motor is used.

Water saving

The velocity of water is less than that in conventional cooling tower. In combination with high efficiency drift eliminators this can reduce the drift loss to 0.001% which is much less than that for a conventional tower. Since the water droplets will be less than 50 micron it evaporates immediately without causing any pollution nearby

Low maintenance cost

Since the fanless cooling tower has no mechanical equipment such as fan, motor, gearbox etc. there is hardly any maintenance required, provided the quality of circulation water is kept clean and well maintained.

7.2 Cooling Tower Performance

The important parameters, from the point of determining the performance of cooling towers, are:

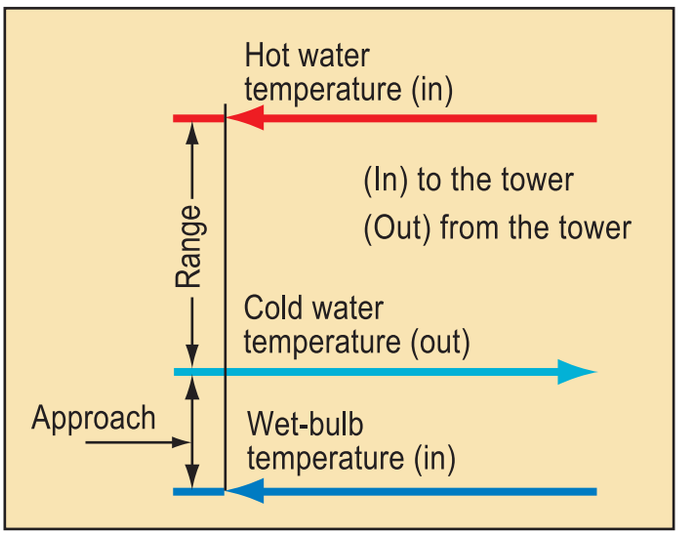


Figure 7.4 Range and Approach

i) "Range" is the difference between the cooling tower water inlet and outlet temperature. (see Figure 7.4).

ii) "Approach" is the difference between the cooling tower outlet cold water temperature and ambient wet bulb temperature. Although, both range and approach should be monitored, the `Approach' is a better indicator of cooling tower performance. (see Figure 7.4).

iii) **Cooling tower effectiveness** (in percentage) is the ratio of range, to the ideal range, i.e., difference between cooling water inlet temperature and ambient wet bulb temperature, or in other words it is = **Range / (Range + Approach).**

iv) Cooling capacity is the heat rejected in kcal/hr or TR, given as product of mass flow rate of water, specific heat and temperature difference.

v) Evaporation loss is the water quantity evaporated for cooling duty and, theoretically, for every 10,00,000 kcal heat rejected, evaporation quantity works out to 1.8 m^3. An empirical relation used often is:

***Evaporation Loss (m^3/hr) = 0.00085 x 1.8 x circulation rate (m^3/hr) x (T1-T2)**

T_1-T_2 = Temp. difference between inlet and outlet water.

*Source: Perry's Chemical Engineers Handbook (Page: 12-17)

vi) Cycles of concentration (C.O.C) is the ratio of dissolved solids in circulating water to the dissolved solids in make up water.

vii) Blow down losses depend upon cycles of concentration and the evaporation losses and is given by relation:

Blow Down = Evaporation Loss / (C.O.C. – 1)

viii) Liquid/Gas (L/G) ratio, of a cooling tower is the ratio between the water and the air mass flow rates. Against design values, seasonal variations require adjustment and tuning of water and air flow rates to get the best cooling tower effectiveness through measures like water box loading changes, blade angle adjustments.

Thermodynamics also dictate that the heat removed from the water must be equal to the heat absorbed by the surrounding air:

$$L(T_1 - T_2) = G(h_2 - h_1)$$

$$\frac{L}{G} = \frac{h_2 - h_1}{T_1 - T_2}$$

Where,

L/G	=	liquid to gas mass flow ratio (kg/kg)
T_1	=	hot water temperature (^{0}C)
T_2	=	cold water temperature (^{0}C)
h_2	=	enthalpy of air-water vapor mixture at exhaust wet-bulb temperature (same units as above)
h_1	=	enthalpy of air-water vapor mixture at inlet wet-bulb temperature (same units as above)

Factors Affecting Cooling Tower Performance

Capacity

Heat dissipation (in kcal/hour) and circulated flow rate (m^3/hr) are not sufficient to understand cooling tower performance. Other factors, which we will see, must be stated along with flow rate m^3/hr. For example, a cooling tower sized to cool 4540 m^3/hr through a 13.9^{o}C range might be larger than a cooling tower to cool 4540 m^3/hr through 19.5^{o}C range.

Range

Range is determined not by the cooling tower, but by the process it is serving. The range at the exchanger is determined entirely by the heat load and the water circulation rate through the exchanger and on to the cooling water.

Range ^{o}C = Heat Load in kcal/hour / Water Circulation Rate in LPH

Thus, Range is a function of the heat load and the flow circulated through the system.

Cooling towers are usually specified to cool a certain flow rate from one temperature to another temperature at a certain wet bulb temperature. For example, the cooling tower might be specified to cool 4540 m^3/hr from 48.9^{o}C to 32.2^{o}C at 26.7^{o}C wet bulb temperature.

Cold Water Temperature 32.2^{o}C – Wet Bulb Temperature (26.7^{o}C) = Approach (5.5^{o}C)

As a generalization, the closer the approach to the wet bulb, the more expensive the cooling tower due to increased size. Usually a 2.8^{o}C approach to the design wet bulb is the coldest water temperature that cooling tower manufacturers will guarantee. If flow rate, range, approach and wet bulb had to be ranked in the order of their importance in sizing a tower, approach would be first with flow rate closely following the range and wet bulb would be of lesser importance.

Factors that affect cooling tower size

Cooling tower size is affected by the heat load, range, approach, and WBT. When three of these four quantities are held constant, tower size varies in the following manner:

- Directly with the heat load
- Inversely with the range
- Inversely with the approach
- Inversely with the entering WBT

Heat Load

The heat load imposed on a cooling tower is determined by the process being served. The degree of cooling required is controlled by the desired operating temperature level of the process. In most cases, a low operating temperature is desirable to increase process efficiency or to improve the quality or quantity of the product. In some applications (e.g. internal combustion engines), however, high operating temperatures are desirable. The size and cost of the cooling tower is proportional to the heat load. If heat load calculations are low undersized equipment will be purchased. If the calculated load is high, oversize and more costly, equipment will result.

Process heat loads may vary considerably depending upon the process involved. Determination of accurate process heat loads can become very complex but proper consideration can produce satisfactory results. On the other hand, air conditioning and refrigeration heat loads can be determined with greater accuracy.

Information is available for the heat rejection requirements of various types of power equipment. A sample list is as follows:

Air Compressor		
- Single-stage	-	129 kcal/kW/hr
- Single-stage with after cooler	-	862 kcal/kW/hr
- Two-stage with intercooler	-	518 kcal/kW/hr
- Two-stage with intercooler and after cooler	-	862 kcal/kW/hr
Refrigeration, Compression	-	63 kcal/min/TR
Refrigeration, Absorption	-	127 kcal/min/TR

Steam Turbine Condenser	-	555 kcal/kg of steam
Diesel Engine, Four-Cycle, Supercharged	-	880 kcal/kW/hr
Natural Gas Engine, Four-cycle (18 kg/cm^2 compression)	-	1523 kcal/kW/hr

Wet Bulb Temperature

Wet bulb temperature is an important factor in performance of evaporative water cooling equipment. It is a controlling factor from the aspect of minimum cold water temperature to which water can be cooled by the evaporative method. Thus, the wet bulb temperature of the air entering the cooling tower determines operating temperature levels throughout the plant, process, or system. Theoretically, a cooling tower will cool water to the entering wet bulb temperature, when operating without a heat load. However, a thermal potential is required to reject heat, so it is not possible to cool water to the entering air wet bulb temperature, when a heat load is applied. The approach obtained is a function of thermal conditions and tower capability.

Initial selection of towers with respect to design wet bulb temperature must be made on the basis of conditions existing at the tower site. The temperature selected is generally close to the average maximum wet bulb for the summer months. An important aspect of wet bulb selection is, whether it is specified as ambient or inlet. The ambient wet bulb is the temperature, which exists generally in the cooling tower area, whereas inlet wet bulb is the wet bulb temperature of the air entering the tower. The later can be, and often is, affected by discharge vapors being recalculated into the tower. Recirculation raises the effective wet bulb temperature of the air entering the tower with corresponding increase in the cold water temperature. Since there is no initial knowledge or control over the recirculation factor, the ambient wet bulb should be specified. The cooling tower supplier is required to furnish a tower of sufficient capability to absorb the effects of the increased wet bulb temperature peculiar to his own equipment.

It is very important to have the cold water temperature low enough to exchange heat or to condense vapours at the optimum temperature level. By evaluating the cost and size of heat exchangers versus the cost and size of the cooling tower, the quantity and temperature of the cooling tower water can be selected to get the maximum economy for the particular process.

The Table 7.1 illustrates the effect of approach on the size and cost of a cooling tower. The towers included were sized to cool 4540 m^3/hr through a 16.67^{o}C range at a 26.7^{o}C design wet bulb. The overall width of all towers is 21.65 meters; the overall height, 15.25 meters, and the pump head, 10.6 m approximately.

Table 7.1 APPROACH Vs. COOLING TOWER SIZE (4540 m^3/hr; 16.67^{o}C Range 26.7^{o}C Wet Bulb; 10.7 m Pump Head)

Approach ^{o}C	2.77	3.33	3.88	4.44	5.0	5.55
Hot Water ^{o}C	46.11	46.66	47.22	47.77	48.3	48.88
Cold Water ^{o}C	29.44	30	30.55	31.11	31.66	32.22
No. of Cells	4	4	3	3	3	3
Length of Cells Mts.	10.98	8.54	10.98	9.76	8.54	8.54
Overall Length Mts.	43.9	34.15	32.93	29.27	25.61	25.61
No. of Fans	4	4	3	3	3	3
Fan Diameter Mts.	7.32	7.32	7.32	7.32	7.32	6.71
Total Fan kW	270	255	240	202.5	183.8	183.8

Approach and Flow

Suppose a cooling tower is installed that is 21.65 m wide × 36.9 m long × 15.24m high, has three 7.32 m diameter fans and each powered by 25 kW motors. The cooling tower cools from 3632 m^3/hr water from 46.1°C to 29.4°C at 26.7°C WBT dissipating 60.69 million kcal/hr. The Table 7.2 shows what would happen with additional flow but with the range remaining constant at 16.67°C. The heat dissipated varies from 60.69 million kcal/hr to 271.3 million kcal/hr.

Table 7.2 Flow vs. Approach for a Given Tower (Tower is 21.65 m × 36.9 M; Three 7.32 M Fans; Three 25 kW Motors; 16.7°C Range with 26.7°C Wet Bulb)

Flow m^3/hr	Approach °C	Cold Water °C	Hot Water °C	Million kcal/hr
3632	2.78	29.40	46.11	60.691
4086	3.33	29.95	46.67	68.318
4563	3.89	30.51	47.22	76.25
5039	4.45	31.07	47.78	84.05
5516	5.00	31.62	48.33	92.17
6060.9	5.56	32.18	48.89	101.28
7150.5	6.67	33.29	50.00	119.48
8736	8.33	35.00	51.67	145.63
11590	11.1	37.80	54.45	191.64
13620	13.9	40.56	57.22	226.91
16276	16.7	43.33	60.00	271.32

For meeting the increased heat load, few modifications would be needed to increase the water flow through the tower. However, at higher capacities, the approach would increase.

Range, Flow and Heat Load

Range is a direct function of the quantity of water circulated and the heat load. Increasing the range as a result of added heat load does require an increase in the tower size. If the cold water temperature is not changed and the range is increased with higher hot water temperature, the driving force between the wet bulb temperature of the air entering the tower and the hot water temperature is increased, the higher level heat is economical to dissipate.

If the hot water temperature is left constant and the range is increased by specifying a lower cold water temperature, the tower size would have to be increased considerably. Not only would the range be increased, but the lower cold water temperature would lower the approach. The resulting change in both range and approach would require a much larger cooling tower.

Approach & Wet Bulb Temperature

The design wet bulb temperature is determined by the geographical location. Usually the design wet bulb temperature selected is not exceeded over 5 percent of the time in that area. Wet bulb temperature is a factor in cooling tower selection; the higher the wet bulb temperature, the smaller the tower required to give a specified approach to the wet bulb at a constant range and flow rate.

A 4540 m^3/hr cooling tower selected for a 16.67°C range and a 4.45°C approach to 21.11°C wet bulb would be larger than a 4540 m^3/hr tower selected for a 16.67°C range and a 4.45°C approach to a 26.67°C wet bulb. Air at the higher wet bulb temperature is capable of picking up more heat. Assume that the wet bulb temperature of the air is increased by approximately 11.1°C. As air removes heat from the water in the tower, each kg of air entering the tower at 21.1°C wet bulb would contain 18.86 kcals and if it were to leave the tower at 32.2°C wet bulb it would contain 24.17 kcal per kg of air.

In the second case, each kg of air entering the tower at 26.67°C wet bulb would contain 24.17 kcal and were to leave at 37.8°C wet bulb it would contain 39.67 kcal per kg of air.

In going from 21.1oC to 32.2oC, 12.1 kcal per kg of air is picked up, while 15.5 kcal/kg of air is picked up in going from 26.67°C to 37.8°C.

Fill Media Effects

In a cooling tower, hot water is distributed above fill media which flows down and is cooled due to evaporation with the intermixing air. Air draft is achieved with use of fans. Thus some power is consumed in pumping the water to a height above the fill and also by fans creating the draft.

An energy efficient or low power consuming cooling tower is to have efficient designs of fill media with appropriate water distribution, drift eliminator, fan, gearbox and motor. Power savings in a cooling tower, with use of efficient fill design, is directly reflected as savings in fan power consumption and pumping head requirement.

Function of Fill media in a Cooling Tower

Heat exchange between air and water is influenced by surface area of heat exchange, time of heat exchange (interaction) and turbulence in water effecting thoroughness of intermixing. Fill media in a cooling tower is responsible to achieve all of above.

Splash and Film Fill Media: As the name indicates, splash fill media generates the required heat exchange area by splashing action of water over fill media and hence breaking into smaller water droplets. Thus, surface of heat exchange is the surface area of the water droplets, which is in contact with air.

Film Fill and its Advantages

In a film fill, water forms a thin film on either side of the fill sheets. Thus area of heat exchange is the surface area of the fill sheets, which is in contact with air.

Typical comparison between various fill media is shown in Table 7.3.

Table 7.3 Typical Comparisons Between Various Fill Media

	Splash Fill	Film Fill	Low Clog Film Fill
Possible L/G Ratio	1.1 – 1.5	1.5 – 2.0	1.4 – 1.8
Effective Heat Exchange Area	30 – 45 m^2/m^3	150 m2/m^3	85 – 100 m^2/m^3
Fill Height Required	5 – 10 m	1.2 – 1.5 m	1.5 – 1.8 m
Pumping Head Requirement	9 – 12 m	5 – 8 m	6 – 9 m
Quantity of Air Required	High	Much low	Low

Due to fewer requirements of air and pumping head, there is a tremendous saving in power with the invention of film fill.

Recently, low-clog film fills with higher flute sizes have been developed to handle high turbid waters. For sea water, low clog film fills are considered as the best choice in terms of power saving and performance compared to conventional splash type fills.

Choosing a Cooling Tower

The counter-flow and cross flows are two basic designs of cooling towers based on the fundamentals of heat exchange. It is well known that counter flow heat exchange is more effective as compared to cross flow or parallel flow heat exchange.

Cross-flow cooling towers are provided with splash fill of concrete, wood or perforated PVC. Counter-flow cooling towers are provided with both film fill and splash fill.

Typical comparison of Cross flow Splash Fill, Counter Flow Tower with Film Fill and Splash fill is shown in Table 7.4. The power consumption is least in Counter Flow Film Fill followed by Counter Flow Splash Fill and Cross-Flow Splash Fill.

Table 7.4 Typical Comparison of Cross flow splash fill, Counter Flow Tower With Film Fill and Splash Fill

Number of Towers	:	2	
Water Flow	:	16000 m^3/hr.	
Hot Water Temperature	:	41.5°C	
Cold Water Temperature	:	32.5°C	
Design Wet Bulb Temperature	:	27.6°C	
	Counter Flow Film Fill	**Counter Flow Splash Fill**	**Cross-Flow Splash Fill**
Fill Height, Meter	1.5	5.2	11.0
Plant Area per Cell	14.4 × 14.4	14.4 × 14.4	12.64 × 5.49
Number of Cells per Tower	6	6	5
Power at Motor Terminal/Tower, kW	253	310	330
Static Pumping Head, Meter	7.2	10.9	12.05

7.3 Efficient System Operation

I Cooling Water Treatment

Cooling water systems is one of the Critical utility in Power plants, process industries and in Air-conditioning systems. The power plant performance, Chiller performance have direct effect on energy consumption, based on Cooling water temperatures which in turn is maintained by good cooling water treatment.

The various problems in Cooling water system and the corrective measures required are discussed below.

a) Water Side Problems

Usually the typical problems that any (Open) cooling system meets with are:

- Corrosion and/or Scale formation
- Biological/Micro-biological fouling

Corrosion:

Corrosion, being not a precisely understood phenomenon, is a function of various factors of which the following are the main factors responsible for promoting corrosion in the system; high salinity of the water, low PH, low Alkalinity, presence of corrosive gases (mainly oxygen and CO_2), dissimilarity of the metals etc.

Corrosion can either lead to failure of the metallurgy (leakages in the heat exchangers) and/or deposit formation of corrosion products.

Scale Formation:

The main sources for the scale formation in the Open Evaporative Condenser circuit are:

Hard water containing, high levels of Calcium and Magnesium, high level of PH and Alkalinity. An open evaporative cooling systems (condenser water systems) operated on softened water can meet with severe scaling problems when

- PH of the circulating water is above 9.0
- The total Alkalinity as $CaCO_3$ is above 550 ppm
- Temporary hardness in the sources of make-up is above 200 ppm

Biological/Micro-Biological Fouling

Systems exposed to sunlight (mainly cooling tower) often meet with severe problem of algae formation. Other problems associated with algae are slime mass, fungi and various species of bacteria.

Bacteria being miniature bodies, of which growth is not controlled, can lead to the formation of fine masses of suspended particles that lead to fouling and deposit formation. Algae obviously block the nozzles of the cooling tower and thus reduce temperature drop across the tower. Slime masses again are responsible for fouling and deposit formation.

Deposit Formation:

Foreign matter such as; turbidity, sand, silt, mud, air borne debris and other suspended impurities are the sources of deposits formation. Corrosion products that are formed also add to the deposit formation.

b) Energy Losses:

Regardless of the type of system, be it open or closed, if it meets with any of the above problems, either the cooling tower nozzles are blocked resulting in reduced Delta 'T' and/or the deposits/scales are formed on the heat transfer surfaces.

For example, the energy losses due to scale and deposit formation in a cooling water circuit of a refrigeration system are significant as shown in Table 7.5. The scale and deposit on the heat transfer area in process equipment can also cause production loss.

Table – 7.5 Effect of Fouling on Efficiency and Power Consumption

Fouling Factor	Thickness of scale/deposit (mm)	% Reduction in efficiency in terms of heat transfer (Condenser)	% Increase in power consumption due to drop in condenser efficiency
Clean	0.00	0	-
0.0005	0.15	30	5
0.001	0.30	44	10
0.002	0.60	63	20
0.003	0.90	72	30

c) Solution to the Problems

ON line / OFF Line Chemical Cleaning

Depending on the criticality the plant management may adopt ON line/ OFF line cleaning systems.

Preventive Treatment

For preventive treatment, a wide range of chemicals are available in the market and formulations manufactured by reputed companies are generally very safe to use in the system.

Corrosion/Scale Inhibitors

To control corrosion and scale formation depending upon the severity of each of the problem, either or both chemicals should be used and the selection of the chemicals should be made in accordance with the quality of the make-up water available for plant operation.

Dispersants (For Deposit Formation)

Suitable dispersants help in controlling the deposit formation and selection of the dispersants is made in accordance with the nature of suspended solids/deposits forming particulate present in the water.

Side Stream Filter

Circulating water having very high levels of turbidity and/or suspended impurities should be facilitated with side stream filters. Side stream filters are generally selected to handle 2% to 5% of the total rate

of circulation, but to ensure that the total water content in the system (hold-up volume) is filtered approximately once in 12 hours.

Bio Dispersants and Biocides

To combat problems arising due to the growth of biological and micro biological species, such as algae, fungi, slime, bacteria etc. It is very essential to select a combination of oxidizing and non-oxidizing biocides. Bio-dispersants are used to remove the upper layer of the biological masses and allow better penetration of biocides in the lower layers of bio-masses.

Chlorination

Chlorination is the most effective and most economical oxidizing biocide. Chlorination for the smaller systems may be done with hypo chlorite based products and for the larger systems having hold-up volume in excess of 100 m^3 be done with suitable gas chlorinators. The safest gas chlorination equipment are vacuum gravity feed type which can be easily installed on either 50 kg or 100 kg chlorine cylinders.

II Drift Loss in the Cooling Towers

It is very difficult to ignore drift problem in cooling towers. Now-a-days most of the end user specification calls for 0.02% drift loss.

With technological development and processing of PVC, manufacturers have brought large change in the drift eliminator shapes and the possibility of making efficient designs of drift eliminators that enable end user to specify the drift loss requirement to as low as 0.003 – 0.001%.

III Cooling Tower Fans

The purpose of a cooling tower fan is to move a specified quantity of air through the system, overcoming the system resistance which is defined as the pressure loss. The product of air flow and the pressure loss is air power developed/work done by the fan; this may be also termed as fan output and input kW depends on fan efficiency.

The fan efficiency in turn is greatly dependent on the profile of the blade. An aerodynamic profile with optimum twist, taper and higher coefficient of lift to coefficient of drop ratio can provide the fan total efficiency as high as 85-92 %. However, this efficiency is drastically affected by the factors such as tip clearance, obstacles to airflow and inlet shape, etc.

As the metallic fans are manufactured by adopting either extrusion or casting process it is always difficult to generate the ideal aerodynamic profiles. The FRP blades are normally hand moulded which facilitates the generation of optimum aerodynamic profile to meet specific duty condition more efficiently. Cases reported where replacement of metallic or Glass fibre reinforced plastic fan blades have been replaced by efficient hollow FRP blades, with resultant fan energy savings of the order of 20-30% and with simple pay back period of 6 to 7 months.

Also, due to lightweight, FRP fans need low starting torque resulting in use of lower HP motors. The lightweight of the fans also increases the life of the gear box, motor and bearing is and allows for easy handling and maintenance.

IV Performance Assessment of Cooling Towers

In operational performance assessment, the typical measurements and observations involved are:

- Cooling tower design data and curves to be referred to as the basis.
- Intake air WBT and DBT at each cell at ground level using a whirling pyschrometer.
- Exhaust air WBT and DBT at each cell using a whirling psychrometer.
- CW inlet temperature at risers or top of tower, using accurate mercury in glass or a digital thermometer.
- CW outlet temperature at full bottom, using accurate mercury in glass or a digital thermometer.
- Process data on heat exchangers, loads on line or power plant control room readings, as relevant.
- CW flow measurements, either direct or inferred from pump motor kW and pump head and flow characteristics.
- CT fan motor amps, volts, kW and blade angle settings
- TDS of cooling water.
- Rated cycles of concentration at the site conditions.
- Observations on nozzle flows, drift eliminators, condition of fills, splash bars, etc.

The findings of one typical trial pertaining to the Cooling Towers of a Thermal Power Plant 3 x 200 MW is given below:

Observations

Unit Load 1 & 3 of the Station	=	398 MW
Mains Frequency	=	49.3
Inlet Cooling Water Temperature ^oC	=	44 (Rated 43^oC)
Outlet Cooling Water Temperature ^oC	=	37.6 (Rated 33^oC)
Air Wet Bulb Temperature near Cell ^oC	=	29.3 (Rated 27.5^oC)
Air Dry Bulb Temperature near Cell ^oC	=	40.8^oC
Number of CT Cells on line with water flow	=	45 (Total 48)
Total Measured Cooling Water Flow m^3/hr	=	70426.76
Measured CT Fan Flow m^3/hr	=	989544

Analysis

CT water Flow/Cell, m^3/hr	=	1565 m^3/hr (1565000 kg/hr) (Rated 1875 m^3/hr)
CT Fan air Flow, m^3/hr (Avg.)	=	989544 m^3/hr (Rated 997200 m^3/hr)
CT Fan air Flow kg/hr (Avg.) @ Density of 1.08 kg/m^3	=	1068708 kg/hr
L/G Ratio of C.T. kg/kg	=	1.46 (Rated 1.74 kg/kg)

CT Range	=	(44 – 37.6) = 6.4°C
CT Approach	=	(37.6 – 29.3) = 8.3°C
% CT Effectiveness	=	$\frac{Range}{(Range + Approach)} \; x \; 100$
	=	$\frac{6.4}{(6.4 + 8.3)} \; x \; 100$
	=	43.53
Rated % CT Effectiveness	=	100 * (43 – 33) / (43 – 27.5)
	=	64.5%
Cooling Duty Handled/Cell in kcal	=	1565 * 6.4 * 10^3
(i.e., Flow * Temperature Difference in kcal/hr)	=	10016 * 10^3 kcal/hr (Rated 18750 * 10^3 kcal/hr)
Evaporation Losses in m^3/hr	=	0.00085 x 1.8 x circulation rate (m^3/hr) x (T_1-T_2)
	=	0.00085 x 1.8 x 1565 x (44-37.6)
	=	15.32 m^3/hr per cell
Percentage Evaporation Loss	=	[15.32/1565]*100
	=	0.97%
Blow down requirement for site COC of 2.7	=	Evaporation losses / (COC–1)
	=	15.32/(2.7–1) per cell i.e., 9.01 m^3/hr
Make up water requirement/cell in m^3/hr	=	Evaporation Loss + Blow down Loss
	=	15.32 + 9.01 = 2433

Comments

- Cooling water flow per cell is much lower, almost by 16.5%, need to investigate CW pump and system performance for improvements. Increasing CW flow through cell was identified as a key result area for improving performance of cooling towers.
- Flow stratification in 3 cooling tower cells identified.
- Algae growth identified in 6 cooling tower cells.
- Cooling tower fans are of GRP type drawing 36.2 kW average. Replacement by efficient hollow FRP fan blades is recommended.

7.4 Flow Control Strategies

Control of tower air flow can be done by varying methods: starting and stopping (ON-OFF) of fans, use of two- or three-speed fan motors, use of automatically adjustable pitch fans, and use of variable speed fans.

ON-OFF fan operation of single speed fans provides the least effective control. Two-speed fans provide better control with further improvement shown with three speed fans. Automatic adjustable pitch fans

and variable-speed fans can provide even closer control of tower cold-water temperature. In multi-cell towers, fans in adjacent cells may be running at different speeds or some may be on and others off depending upon the tower load and required water temperature. Depending upon the method of air volume control selected, control strategies can be determined to minimise fan energy while achieving the desired control of the Cold water temperature.

7.5 Energy Saving Opportunities in Cooling Towers

- Follow manufacturer's recommended clearances around cooling towers and relocate or modify structures that interfere with the air intake or exhaust.
- Optimise cooling tower fan blade angle on a seasonal and/or load basis.
- Correct excessive and/or uneven fan blade tip clearance and poor fan balance.
- On old counter-flow cooling towers, replace old spray type nozzles with new square spray ABS practically non-clogging nozzles.
- Replace splash bars with self-extinguishing PVC cellular film fill.
- Install new nozzles to obtain a more uniform water pattern
- Periodically clean plugged cooling tower distribution nozzles.
- Balance flow to cooling tower hot water basins.
- Cover hot water basins to minimise algae growth that contributes to fouling.
- Optimise blow down flow rate, as per COC limit.
- Replace slat type drift eliminators with low pressure drop, self extinguishing, PVC cellular units.
- Restrict flows through large loads to design values.
- Segregate high heat loads like furnaces, air compressors, DG sets, and isolate cooling towers for sensitive applications like A/C plants, condensers of captive power plant etc. A 1^{o}C cooling water temperature increase may increase A/C compressor kW by 2.7%. A 1^{o}C drop in cooling water temperature can give a heat rate savings of 5 kcal/kWh in a thermal power plant.
- Monitor L/G ratio, CW flow rates w.r.t. design as well as seasonal variations. It would help to increase water load during summer and times when approach is high and increase air flow during monsoon times and when approach is narrow.
- Monitor approach, effectiveness and cooling capacity for continuous optimisation efforts, as per seasonal variations as well as load side variations.
- Consider COC improvement measures for water savings.
- Consider energy efficient FRP blade adoption for fan energy savings.
- Consider possible improvements on CW pumps w.r.t. efficiency improvement.
- Control cooling tower fans based on leaving water temperatures especially in case of small units.

– Optimise process CW flow requirements, to save on pumping energy, cooling load, evaporation losses (directly proportional to circulation rate) and blow down losses.

7.6 Case Study: Application of VFD for Cooling Tower (CT) Fan

The rating (KW) of the CT fan is selected for the worst case wet and dry bulb temperatures. In areas where such temperature conditions occur for a small portion of the year & which require maximum air flow for this condition, it is possible to improve energy efficiency by reducing the speed of the fan (to obtain reduced air flow), using a VFD.

It is therefore necessary to obtain data for the variations in wet and dry bulb temperatures on an annual basis to arrive at estimates for the energy saved through use of VFD. Alternatively, it is also possible to install a VFD on a trial basis on the CT fans and measure the electrical power consumed with and without VFD.

The relationship between the power consumed by the CT fan and the airflow delivered by it follows a cube law. The potential for energy savings exists if a proper analysis of the cooling system is made.

Implementation with VFD

An energy efficient system with VFD can be realized through the use of closed loop control. In this control method, the return or cold water temperature is used as the feedback signal to the PID controller which is a standard control block in the drive.

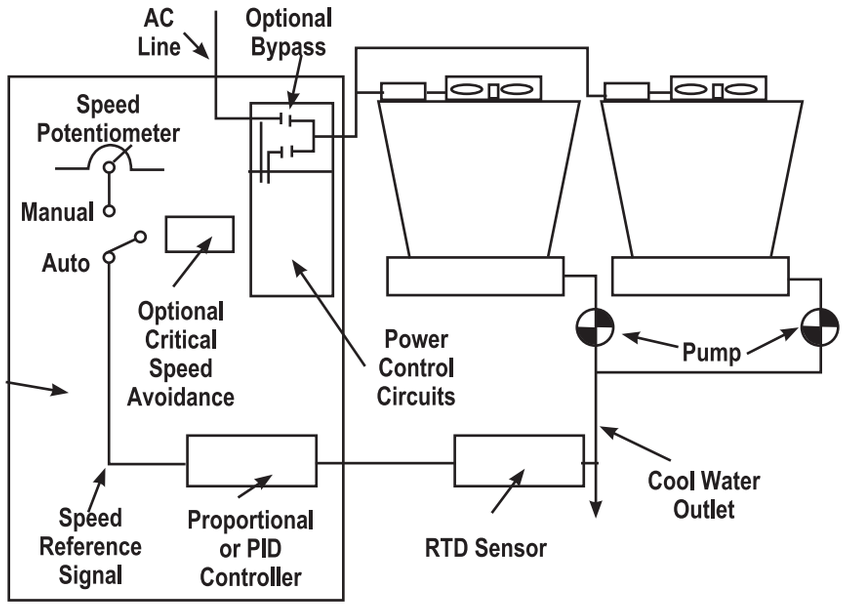


Figure 7.5 Schematic of CT fan control with VFD & closed loop feedback

The highlights of control with temperature feedback and drive can be summarized below:

- An RTD sensor, installed at the CT outlet generates a 4-20 mA current signal as the feedback to the integrated Process PID Controller in the drive.

- The set point for the cooled water temperature is entered in engineering units (°C or °F) in the drive controller.
- Any error between the set point and the feedback signal (temperature in this case) will be integrated by the PID controller so that the same, after correction, is zero. For example, if there is an increase in the outlet temperature (due to wet bulb temperature increase or due to an increase in plant load), the feedback exceeds the set point & the error Δ becomes negative. The PID controller output will now try to increase the drive frequency so that the fans deliver more cooling air for evaporation. This has the effect of bringing down the outlet temperature. The correction continues till the feedback signal matches the set point. A similar correction takes place when the outlet temperature reduces. In that case, the CT fan motor speed is reduced to bring the Δ value to zero.
- *This design therefore permits precise control of outlet temperature and conserves energy.*
- In the event of drive failure, the CT fans can still be operated through an optional built-in bypass circuit which will transfer the power source to the mains supply, thereby ensuring uninterrupted operation.

Use of VFD for CT fan motors in Ingot manufacturing plant

An aluminium ingot manufacturing plant requires large amounts of water for cooling of the ingots. Hence cooling tower fans are required to cool the water from the ingot plant. The salient features of the application are as given below:

- Drives have been installed on two Cooling Towers.
- Details of drives supplied as follows:

Cooling tower (CT)	No of CT fans	CT fan motor kW	Drive kW
1	3	11	37
2	3	15	45

Details of control and power consumption

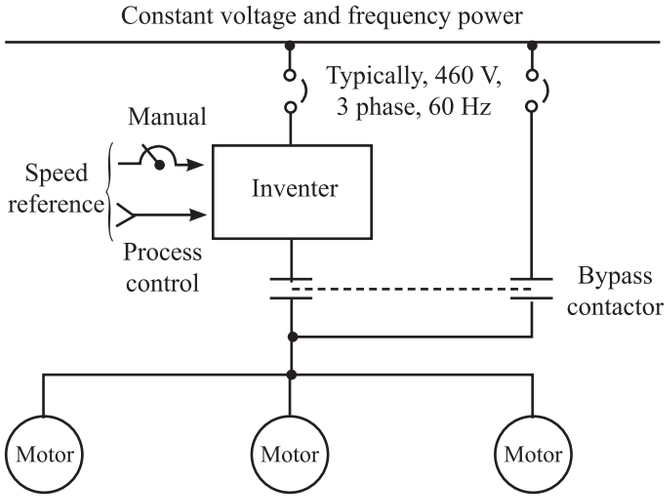
The previous method of control employed a digital temperature controller to switch ON & OFF the CT Fans depending upon the basin temperature. A typical daily regimen employed for CT#1 was as follows:

(a) 3 Nos. CT Fans running for 8 Hours.

(b) 2 Nos. CT Fans running for 8 Hours.

(c) 1 No CT Fan running for 5 hours.

Trials were taken with the VFD (common to 3 nos. drive motors as shown in Figure 7.6 below) installed in CT 1 and run for a period of one month to ascertain the power consumption with and without the drive.



Note:

- The applicable power supply voltage in this case would be 415V, 50Hz.
- There is a bypass line which will enable the motors to operate from mains supply in case of VFD failure.

Figure 7.6 Typical Power Schematic of VFD and CT fan motors

It was compared with the power consumed with manual operation (as described above) on a daily basis. It yielded a significant result in terms of power saved daily.

Savings calculation with VFD operation:

Energy consumed daily with manual control (kWh)	=	392
Energy consumed daily with VFD control (kWh)	=	254
Energy saved on daily basis with VFD control (kWh)	=	138
Energy saved on daily basis with VFD control (%)	=	35.2
Unit energy cost (INR)	=	5
Number of days of running of fans in a month	=	26
Number of months in a year	=	12
Annual savings due to VFD operation (INR)	=	215280
Average price of drive panel (INR)	=	250000
Payback period (Year)	=	1.16

*Data for savings extrapolated for annual estimates

In practice, the power saved with VFD operation would also depend upon the wet bulb temperature which would vary on a seasonal basis. In case of higher temperatures, the VFD would be required to run at maximum speed during which period, the savings would be negligible. Hence the average annual savings would reduce depending upon the site environmental conditions. The quantum of savings can be optimized by having a closed loop system as shown above which will track the outlet water temperature and determine the drive motor speed accurately.

Solved Example:

The energy audit observations at a cooling tower (CT) in a process industry are given below:

Cooling Water (CW) Flow: 3000 m^3/hr

CW in Temperature: 41deg. C

CW Out Temperature: 31 deg C

Wet Bulb Temperature: 24 deg. C

Find out Range, Approach, Effectiveness and cooling tower capacity in kcal per hour of the CT?

Ans:

Range = (Inlet -Outlet) Cooling Water Temperature deg. C

Range = (41 – 31) = 10 deg. C

Approach = (Outlet Cooling Water - Air Wet Bulb) Temperature deg. C

Approach = (31 – 24) = 7 deg C

% CT Effectiveness = Range / (Range + Approach) x 100

% Effectiveness = 100 x [Range / (Approach + Range)]

= 10 / [10+7] x 100 = 58.8 %

Cooling capacity, kcal/hr = heat rejected = CW flow rate in kg per hour x (CW inlet hot water temp. to CT, deg. C - CW outlet cold well temp., deg. C)

Cooling capacity = 3000X1000X (41 - 31) = 30,000,000 kCal per hour = 30 Million kcal/hour.

QUESTIONS	
Objective Type Questions	
1.	The ratio of dissolved solids in circulating water to the dissolved solids in make up water is termed as. a) liquid gas ratio b) cycles of concentration c) cooling tower effectiveness d) none of the above
2.	If inlet and outlet water temperatures of a cooling tower are 40°C and 32°C respectively and atmospheric DBT and WBT are 35 °C and 28 °C respectively then the approach of cooling tower is a) 3°C b) 4°C c) 5°C d) 7°C
3.	Higher the COC in cooling tower , the blow down quantity will a) increases b) decreases c) no change d) none of the above
4.	The L/G ratio of a cooling tower does not depend on a) range b) enthalpy of inlet air c) outlet wet bulb temperature d) dry bulb temperature
5.	Which of the following ambient conditions will evaporate minimum amount of water in a cooling tower a) 35°C DBT and 30°C WBT b) 38°C DBT and 31°C WBT c) 38°C DBT and 37°C WBT d) 35°C DBT and 29°C WBT
6.	Natural draft cooling towers are mainly used in ____. a) steel industry b) alumina industry c) fertilizer industry d) power stations
7.	Cooling tower effectiveness is the ratio of --------- a) range / (range + approach) b) approach/(range + approach) c) range/approach d) approach/range
8.	Which one of the following has maximum effect on cooling tower performance: a) fill media b) drift c) louvers d) casing
9.	Normally the guaranteed best approach a cooling tower can achieve is a) 5°C b) 8°C c) 12°C d) 2.8 °C
10.	Select the wrong statement --- a) for a given heat rejection duty, a higher range will reduce the circulating water flow rate b) when the cycle of concentration is left at one, all water left in the cooling tower after evaporation needs to be removed as blowdown. c) a better indicators for cooling tower performance is Range d) a cooling tower size will be greater for 20°C Wet bulb temperature (WBT) than for a 30°C WBT, for the same circulation, range and approach

	Short Type Questions
S-1	Estimate the cooling tower capacity (TR) with the following parameters Water flow rate through CT = 120 m^3/h Sp. heat of water = 1 kcal/kg °C Inlet water temperature = 37 °C Outlet water temperature = 32 °C Ambient WBT = 29 °C
S-2	What do you mean by the term Cycles of Concentration and how it is related to cooling tower blow down?
S-3	In a cooling tower, the cooling water circulation rate is 1250 m^3/hr. The operating range is 7°C. If the blowdown rate of the cooling tower is 1 % of the circulation rate, calculate the evaporation loss and COC
S-4	Explain the term L/G ratio?
S-5	What is the function of fill media in a cooling tower?
	Long Type Questions
L-1	A Cooling Tower cools 1565 m3/hr of water from 44° C to 37.6° C at 29.3° C wet bulb temperature. The cooling tower fan flow air rate is 989544 m^3/hr (air density =1.08 kg/m^3) and operates at 2.7 cycles of concentration. Find out the following a) Range, b) Approach, c) % CT Effectiveness d) L/G Ratio in kg/kg e) Cooling Duty Handled in TR f) Evaporation Losses in m^3/hr g) Blow down requirement in m^3/hr h) Make up water requirement/cell in m^3/hr
L-2	Explain the significance of the following on cooling tower performance a) Range, b) Approach, c) Effectiveness d) L/G Ratio e) WBT

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1. ASHRAE Handbook
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3. Inputs on VFD for CT fans: Danfoss India Pvt. Ltd.

Chapter 8

Lighting System

Learning Objectives

In this chapter you will learn about

- Light sources and Types of lamps
- Basic parameters and terminologies used in lighting systems
- Recommended illuminance standards
- Method of calculating illuminance
- Energy efficient lighting controls
- Standards and labeling programme for FTL lamps

8. LIGHTING SYSTEM

Light Source, T5, T8, T12 Definitions, CFL, LEDs, Metal Halides, Choice of Lighting, Luminance Requirements and Energy Conservation Avenues, Electronic Ballast, Energy Efficient Lighting Controls, Occupancy Sensors, High Efficiency Street Lighting, Labeling Scheme.

8.1 Introduction

Most natural light comes from the sun, including moon light. Its origin makes it completely clean and it consumes no natural resources. But man-made sources generally require consumption of resources, such as fossil fuels, to convert stored energy into light energy.

Light is usually described as the type of electromagnetic radiation that has a wavelength visible to the human eye, roughly 400 to 700 nanometers. Light exists as tiny "packets" called photons and exhibits the properties of both particles and waves. Visible light, as can be seen on the electromagnetic spectrum, as given in Figure 8.1, represents a narrow band between ultraviolet light (UV) and infrared energy (heat). These light waves are capable of exciting the eye's retina, which results in a visual sensation called sight. Therefore, seeing requires a functioning eye and visible light.

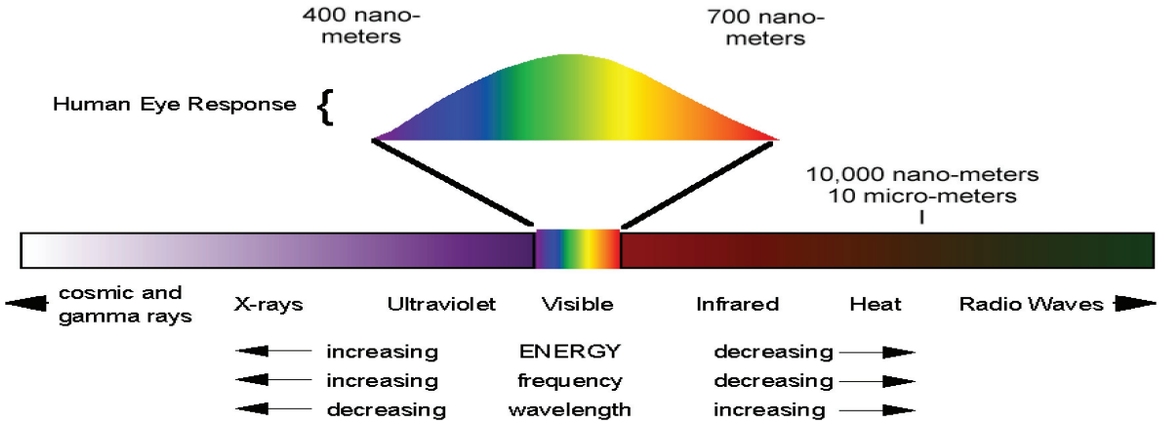


Figure 8.1 Visible Radiation

The lumen (lm) is the photometric equivalent of the Watt, weighted to match the eye response of the "standard observer". Yellowish-green light receives the greatest weight because it stimulates the eye more than blue or red light of equal radiometric power:

1 Watt = 683 lumens at 555 nm wavelength

The best eye sensitivity, as seen from Figure 8.2 is at 555 nm wavelength having greenish yellow colour with a luminous efficacy of 683 lm/Watt.

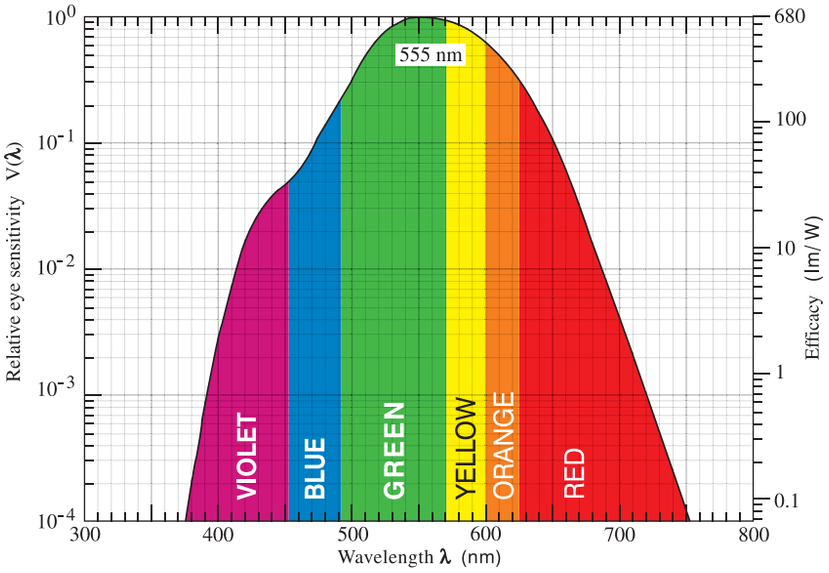


Figure 8.2 Relative Eye Sensitivity and Luminous Efficacy

Three primary considerations to ensure energy efficiency in lighting systems are:

i. Selection of the most efficient light source possible in order to minimize electricity consumption and cost.

ii. Matching the proper lamp type to the intended work task or aesthetic application, consistent with color, brightness control and other requirements.

iii. Establishing adequate light levels without compromising productivity improve security and increase safety.

8.2 Basic Parameters and Terms in Lighting System

Luminous flux: The luminous flux describes the quantity of light emitted by a light source. It is a measure of a lamp's economic efficiency.

The most common measurement or unit of luminous flux is the **lumen (lm)**. The lumen rating of a lamp is a measure of the total light output of the lamp. Light sources are labeled with an output rating in lumens.

Illuminance (E): is the quotient of the luminous flux incident on an element of the surface at a point of surface containing the point, by the area of that element.

The lighting level produced by a lighting installation is usually qualified by the illuminance produced on a specified plane. In most cases, this plane is the major plane of the tasks carried out in the interior and is commonly called the working plane. The illuminance provided by an installation affects both the performance of the tasks and the appearance of the space. **Lux (lx) is** the metric unit of measure for illuminance of a surface. One lux is equal to one lumen per square meter. Illuminance decreases by the square of the distance (inverse square law).

The inverse square law defines the relationship between the illuminance from a point source and distance. It states that the intensity of light per unit area is inversely proportional to the square of the distance from the source (essentially the radius).

$$\mathbf{E = \frac{I}{d^2}}$$

Where, E = Illuminance in lux (lm/m^2), I = Luminous flux in lumen (lm) and d = distance in m

An alternate form of this equation which is sometimes more convenient is:

$$\boldsymbol{E_1 \, d_1^{\,2} = E_2 \, d_2^{\,2}}$$

Distance is measured from the test point to the first luminating surface - the filament of a clear bulb or the glass envelope of a frosted bulb.

Example:

The illuminance is 10 lm/m² from a lamp at 1 meter distance. What will be the illuminance at half the distance?

Solution:

$$E_{(1m)} = (d_2 / d_1)^2 * E_2$$
$$= (1.0 / 0.5)^2 * 10.0$$
$$= 40 \text{ lm/m}^2$$

Average maintained illuminance: is the average of illuminance (lux) levels measured at various points in a defined area.

Circuit Watts: is the total power drawn by lamps and ballasts in a lighting circuit under assessment.

Luminous Efficacy (lm/W): is the ratio of luminous flux emitted by a lamp to the power consumed by the lamp. It is a reflection of efficiency of energy conversion from electricity to light form. **Unit: lumens per lamp Watt (lm/W).**

Lamp Circuit Efficacy: is the amount of light (lumens) emitted by a lamp for each Watt of power consumed by the lamp circuit, i.e. including control gear losses. This is a more meaningful measure for those lamps that require control gear. **Unit: lumens per circuit Watt (lm/W).**

Installed Load Efficacy: is the average maintained illuminance provided on a horizontal working plane per circuit watt with general lighting of an interior. **Unit: lux per Watt per square metre (lux/W/m²).**

Installed Power Density: The installed power density per 100 lux is the power needed per square metre of floor area to achieve 100 lux of average maintained illuminance on a horizontal working plane with general lighting of an interior. **Unit: Watts per square metre per 100 lux (W/m²/100 lux)**

Color rendering index (CRI): is a measure of the effect of light on the perceived color of objects. To determine the CRI of a lamp, the color appearances of a set of standard color chips are measured with special equipment under a reference light source with the same correlated color temperature as the lamp being evaluated. If the lamp renders the color of the chips identical to the reference light source, its CRI is 100. If the color rendering differs from the reference light source, the CRI is less than 100. A low CRI indicates that some colors may appear unnatural when illuminated by the lamp.

Luminaire: is a device that distributes filters or transforms the light emitted from one or more lamps. The luminaire includes all the parts necessary for fixing and protecting the lamps, except the lamps themselves. In some cases, luminaires also include the necessary circuit auxiliaries, together with the means for connecting them to the electric supply. The basic physical principles used in optical luminaire are reflection, absorption, transmission and refraction.

Control gear: The gears used in the lighting equipment are as follows:

- **Ballast** is a current limiting device, to counter negative resistance characteristics of any discharge lamps. In case of fluorescent lamps, it aids the initial voltage build-up, required for starting. In an electric circuit the ballast acts as a stabilizer. Fluorescent lamp is basically an electric discharge lamp with two electrodes separated inside a tube with no apparent connection between them. When sufficient voltage is impressed on these electrodes, electrons are driven from one electrode and attracted to the other. The current flow takes place through an atmosphere of low-pressure mercury vapour.
- Since the fluorescent lamps cannot produce light by direct connection to the power source, they need an ancillary circuit and device to get started and remain illuminated. The auxiliary circuit housed in a casing is known as ballast.
- **Ignitors** are used for starting high intensity discharge lamps such as metal halide and sodium vapour lamps. Ignitors generate a high voltage pulse or a series of pulses to initiate the discharge.

8.3 Light Source and Lamp Types

Lamp is equipment, which produces light. Light is that part of the electromagnetic spectrum that is perceived by our eyes. A number of light sources are available, each with its own unique combination of operating characteristics viz., efficacy, colour, lamp life, and the percent of output that a lamp loses over its life.

Based on the construction and operating characteristics, the lamps can be categorized into three groups: incandescent, fluorescent and high intensity discharge (HID) lamps. HID lamps can be further classified as sodium vapour, mercury vapour and metal halide lamps. The most commonly used lamps are described briefly as follows:

1) Incandescent lamp

The principal parts of an incandescent lamp also known as GLS lamp (General Lighting Service lamp) include the filament, the bulb, the fill gas or vacuum and the cap. Incandescent lamps (Figure 8.3 A&B) produce light by means of a wire or filament heated to incandescence by the flow of electric current

through it. The filament is enclosed in an evacuated glass bulb filled with inert gas such as argon, krypton, or nitrogen that helps to increase the brilliance of lamp and to prevent the filament from burning out.

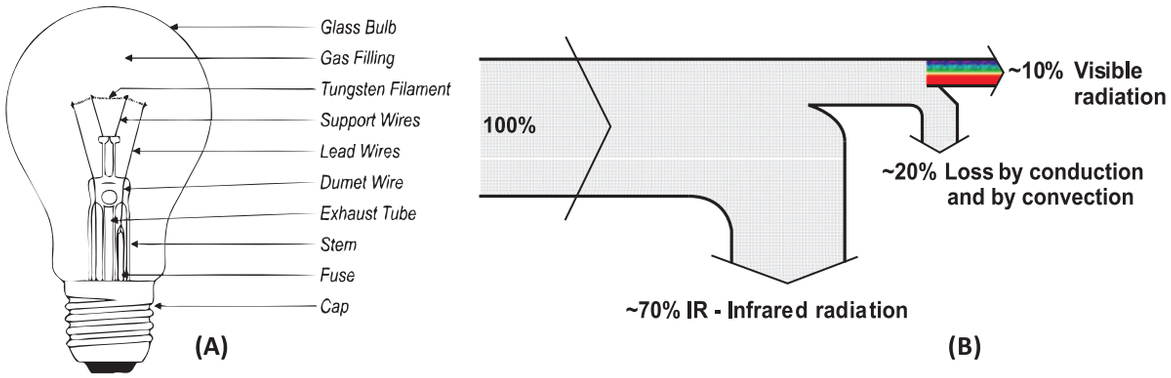


Figure 8.3 Incandescent Lamp and Energy Flow Diagram

Reflector lamps: Reflector lamps are basically incandescent, provided with a high quality internal mirror, which follows exactly the parabolic shape of the lamp. The reflector is resistant to corrosion, thus making the lamp maintenance free and output efficient.

2) Halogen lamp

It has a tungsten filament and the bulb filled with halogen gas (Figure 8.4). Current flows through the filament and heats it up, as in incandescent lamps. These lamps therefore generate a relatively large amount of heat. The use of halogen increases the efficiency and extends the service life compared with traditional incandescent lamps. Low-voltage types are very small and are ideal for precise direction of light, but they require a transformer.

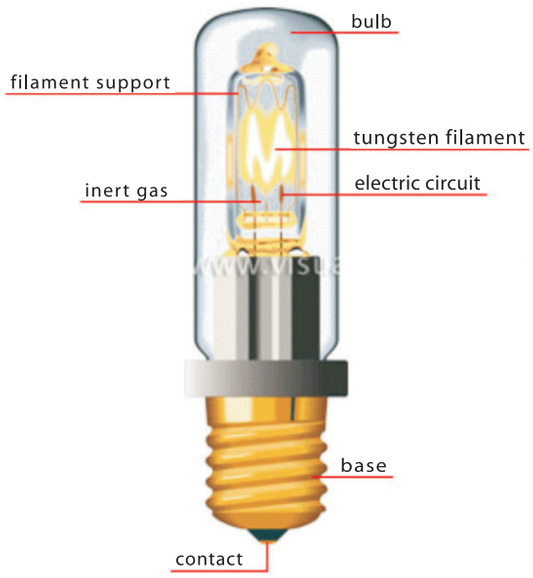


Figure 8.4 Halogen Lamp

Tungsten atoms evaporate from the hot filament and move toward the cooler wall of the bulb. Tungsten, oxygen and halogen atoms combine at the bulb-wall to form tungsten oxyhalide molecules. The bulb-wall temperature keeps the tungsten oxyhalide molecules in a vapor. The molecules move toward the hot filament where the higher temperature breaks them apart. Tungsten atoms are re-deposited on the cooler regions of the filament - not in the exact places from which they evaporated. Breaks usually occur near the connections between the tungsten filament and its molybdenum lead-in wires where the temperature drops sharply.

3) Fluorescent tube lamp (FTL)

It works by the fluorescence principle. A fluorescent lamp (Figure 8.5 A&B) is a glass tube containing a small trace of a gas such as mercury vapor (for a white color), carbon dioxide (for green), neon (for red color), etc., with a special fluorescent / phosphorescent coating on the interior surface of the tube. It contains two filaments, one at each end of the tube and when the electrical supply is switched ON, the contacts of the starter open and the filaments glow to heat up the gas contained inside the tube.

This action provides a voltage across its electrodes that set off an electric (gaseous mercury) arc discharge in the tube. This generates invisible UV radiation that is high enough to ionise the warmed-up gas inside the tube. This ionised gas also called as "plasma", excites the fluorescent coating so that it gives out visible light. Ballast is needed to start and operate fluorescent lamps, because of the characteristics of a gaseous arc. The luminous flux is highly dependent on the ambient temperature. Fluorescent Lamps are about 3 to 5 times as efficient as standard incandescent lamps and can last about 10 to 20 times longer.

The different types of fluorescent lamps and their reference are given below:

Linear tubes

- T12 - 38 mm (1.5"diameter)
- T8 - 25 mm (1"diameter)
- T5 - 16mm (5/8"diameter)
- T2 - 6 mm (1/4''diameter)

U-bent tubes

- T12- 38 mm (1.5"diameter)
- T8 - 25 mm (1" diameter)

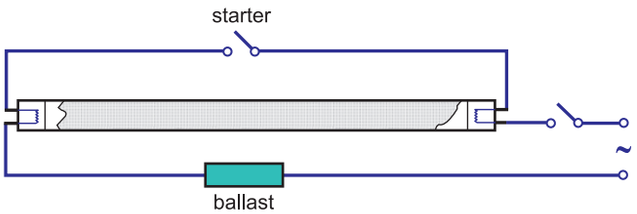


Figure 8.5 (A) Fluorescent Tube Lamp

Circular tubes

- T9 - 38 mm (1.5"diameter)
- T5 - 16 mm (5/8"diameter)

These four lamps vary in diameter (ranging from 1.5 inches that is 12/8 of an inch for T12 to 0.625 or 5/8 of an inch in diameter for T5 lamps). Efficacy is another area that distinguishes one from another. T5 & T8 lamps offer a 5-percent increase in efficacy over 40-watt T12 lamps, and have become the most popular choice for new installations.

4) Compact fluorescent lamp (CFL)

Compact Fluorescent lamps (Figure 8.6) are compact / miniature versions of the linear or circular fluorescent lamps and operate in a very similar way. The luminous flux depends on temperature. CFL's use less power and have a longer rated life compared to an incandescent lamp.

Figure 8.6 Compact Fluorescent Lamp

They are designed to replace an incandescent lamp and can fit into most existing light fixtures formerly used for incandescent. CFL's are available in screw type/ pin type which fit into standard sockets, and gives off light that is similar to common fluorescent lamps.

5) Sodium vapour lamp

Low pressure sodium vapour lamp

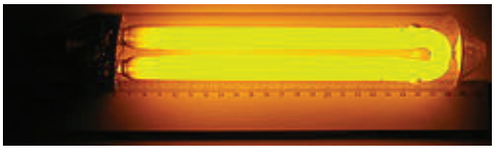


Figure 8.7 Low Pressure Sodium Vapour Lamp

Although low pressure sodium vapour (LPSV) lamps (Figure 8.7) are similar to fluorescent systems (because they are low pressure systems), they are commonly included in the HID family. LPSV lamps are the most efficacious light sources, but they produce the poorest quality light of all the lamp types. Being a monochromatic light source, all colours appear black, white, or shades of gray under an LPSV source. LPSV lamps are available in wattages ranging from 18-180.

LPSV lamp use has been generally limited to outdoor applications such as security or street lighting and indoor, low-wattage applications where color quality is not important (e.g. stairwells). However, because the color rendition is so poor, many municipalities do not allow them for roadway lighting.

High pressure sodium vapour lamp

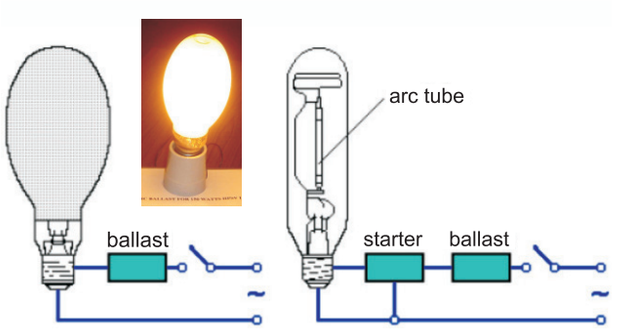


Figure 8.8 (A) High Pressure Sodium Vapour Lamp

The high pressure sodium vapour (HPSV) lamp (Figure 8.8 A&B) is widely used for outdoor and industrial applications as the light is yellowish. Its higher efficacy makes it a better choice than metal halide for these applications, especially when good color rendering is not a priority. HPSV lamps differ from mercury and metal-halide lamps in that they do not contain starting electrodes; the ballast circuit includes a high-voltage electronic starter. The arc tube is made of a ceramic material, which can withstand temperatures up to 1300 °C. It is filled with xenon to help start the arc, as well as a sodium-mercury gas mixture.

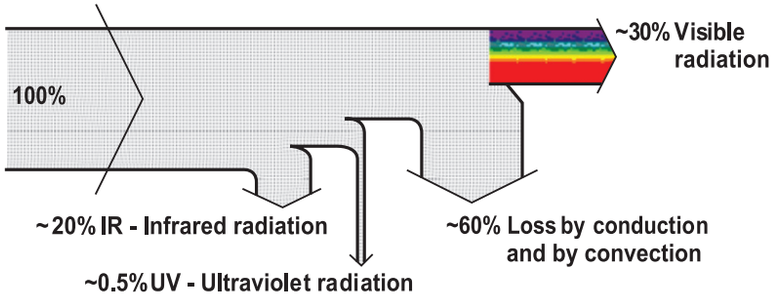


Figure 8.8 (B) Energy Flow diagram of High Pressure Sodium Vapour Lamp

6) Mercury vapour lamp

In a mercury vapour lamp (Figure 8.9) electromagnetic radiation is created from discharge within mercury vapour, but the regime is different than that found in the normal fluorescent lamp. During operation, the pressure within the lamp is in the range of 200 – 400 kPa (compared with only 1 Pa). It is not possible to achieve the mercury vapour discharge in a cold lamp. For this reason, the lamp also includes argon, and the initial arc is struck as an argon arc. The energy from this discharge vapourises the mercury to get the main discharge going.

The mercury vapour lamp produces a much greater proportion of visible light than fluorescent lamp and gives off a bluish white light. Phosphor coating can be given to improve the colour rendering index.

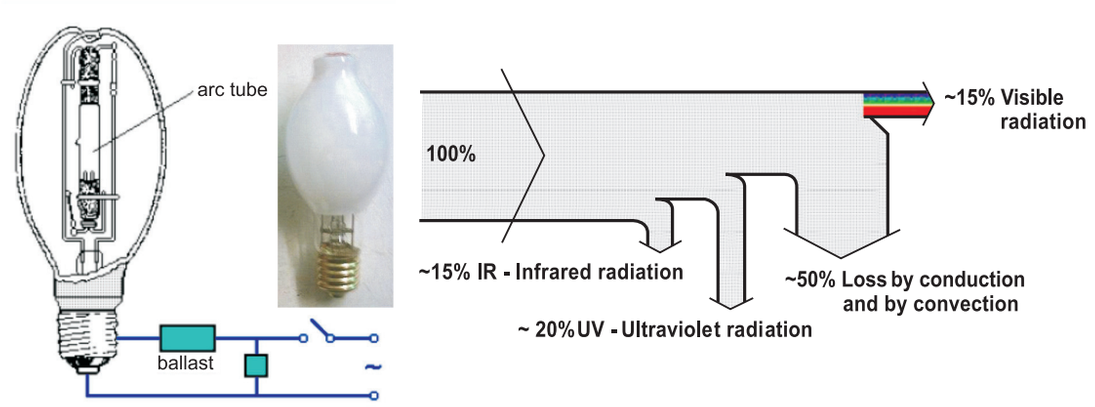


Figure 8.9 Mercury Vapour Lamp and Energy Flow Diagram

7) Metal halide lamp:

Metal halide lamp (Figure 8.10 A&B) can be considered as a variant of high pressure mercury vapour lamp (HPMV). In addition to mercury vapour and argon, this lamp contains metal halide. The halides can be a mixture of rare earth halides, usually iodides or a mixture of sodium and scandium iodide. The mercury vapour radiation is augmented by that of the metals.

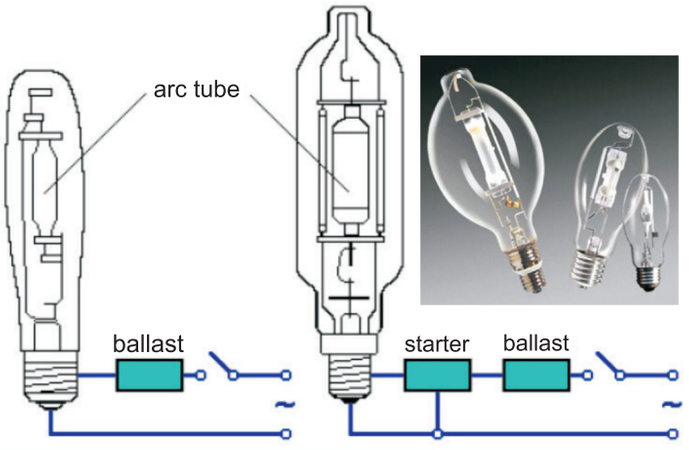


Figure 8.10 (A) Metal Halide Lamp

A highly compact electric arc is produced in a discharge tube. A starter is needed to switch on the lamp. The use of ceramic discharge tubes further improves the lamp properties. The halides act in a similar manner to the tungsten halogen cycle. As the temperature increases there is disassociation of the halide compound releasing the metal into the arc. The halides prevent the quartz wall getting attacked by the alkali metals. By adding other metals to the mercury

different spectrum can be emitted. Some lamps use a third electrode for starting, but others, especially the smaller display lamps, require a high voltage ignition pulse.

Metal halide lamps have a significantly better colour rendering index than mercury vapour and can be tailored by the choice of halides.

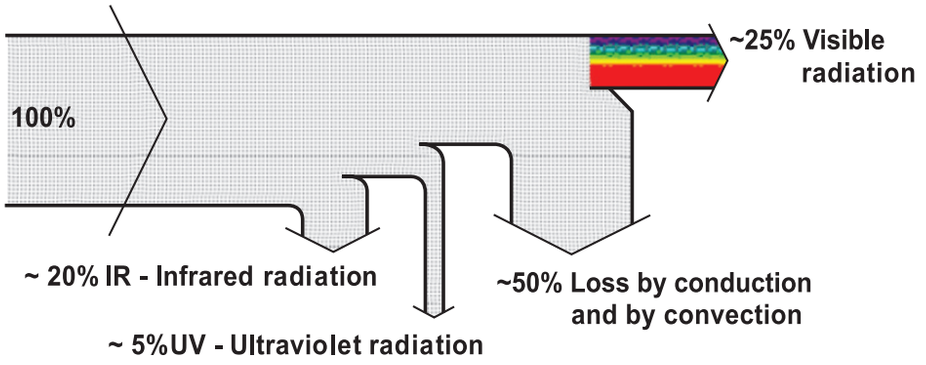


Figure 8.10 (B) Energy Flow Diagram of Metal Halide Lamp

8) Light emitting diode (LED) lamp

LEDs produce light in a very unique way; they produce light via a process called electro-luminescence (Figure 8.11), a process that starts by turning a semiconductor material into a conducting material. A semiconductor with extra electrons is called N-type (negative) material, since it has extra negatively-charged electrons. In N-type material, free electrons can move from a negatively-charged area to a positively charged area. A semiconductor with extra holes is called P-type (positive) material since it has extra positively-charged gaps called holes. When excited with current the negative electron leaves its atom and the P-type material's positive attraction draws the free negative electron into its hole, and the hole also moves toward the electron, so on and so forth.

As an electron travels to a hole, it carries energy, but in order to fit into the hole it must release any extra energy, and when it does, the extra energy is released in the form of light. When we maintain a steady flow of electrical current to the diode, it continues the process of allowing electrons to flow from the negative charged material and fall into the positive charged holes which maintains a steady stream of light out of the LED. The actual LED is quite small in size, usually less than one square millimeter. Additional optical components are added to shape and direct the light. LED's are made of number of inorganic semiconductor materials, many of which produce different colour of light.

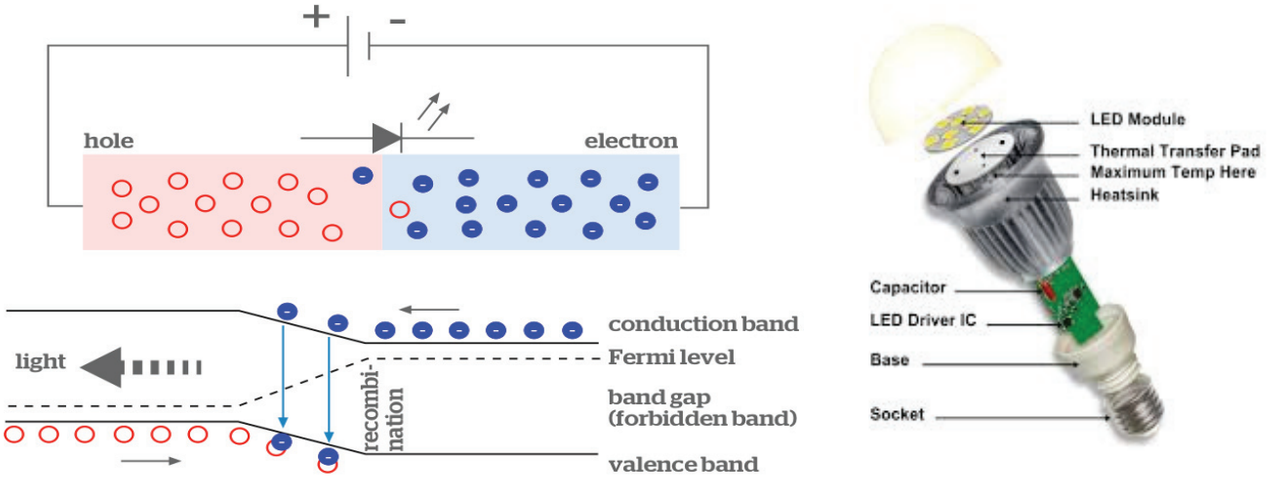


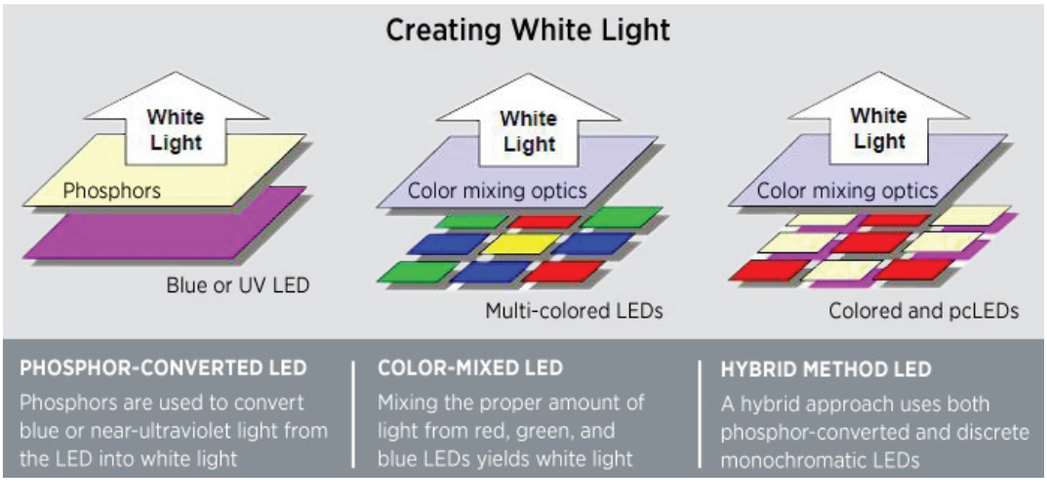
Figure 8.11(A) Representation of LED light

The efficiency of LED's has now risen sharply and is currently up to 200 lumens per watt in the laboratory and in some products available on the market (although more typical LED's average output varies from 50 to 130 lumens per watt).

Because of the low power requirement for LED's, using solar panels becomes more practical and less expensive than running an electrical wire or using a generator. Hence LED with battery backup for remote application is very economical. They do not radiate light in 360 degrees as an incandescent does. The light will be bright wherever it is focused.

Unlike incandescent and fluorescent lamps, LEDs are not inherently white light sources. Instead, LEDs emit nearly monochromatic light, making them highly efficient for colored light applications such as traffic lights and exit signs. However, to be used as a general light source, white light is needed. White light can be achieved with LEDs in three ways:

Phosphor conversion, in which a phosphor is used on or near the LED to convert the colored light to white light; RGB systems, in which light from multiple monochromatic LEDs (red, green, and blue) is mixed, resulting in white light; and a hybrid method, which uses both phosphor-converted and monochromatic LEDs.



Advantages of LED technology is as follows Low power consumption, Directional light output, High efficiency level, Long life: upto ~100,000 hour life if junction temperature can be controlled, Instant switching on with no warm up time, High resistance to switching cycles, High impact and vibration resistance, No UV or IR radiation, Color control ability, allows dimming and Mercury free

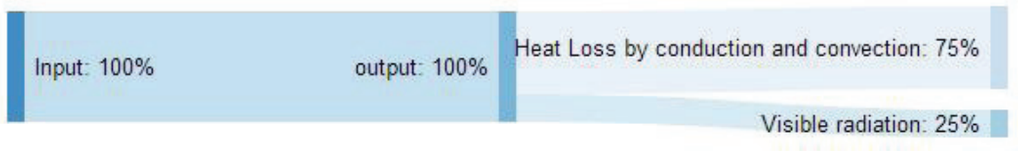


Figure 8.11 (B) Energy Flow diagram of LED Lamp

LED's also offer a number of promising environmental benefits, and they are often viewed as the future of green lighting.

9) Induction lamp

Induction lamp is noted for 'crisp white light output'. Uses a magnetic field to excite gases – has no lamp parts to wear out. It consists of two main components: ballast and a sealed gas-filled bulb. Light is produced via electromagnetic induction, without an electrode or any electrical connection inside the bulb. Instead, high frequency electromagnetic fields are induced from outside the sealed chamber.

To produce light, the ballast supplies the electric coils with high frequency electrical current. The ferrite magnets on either side of the bulb then emit electromagnetic fields which excite electrons within the bulb.

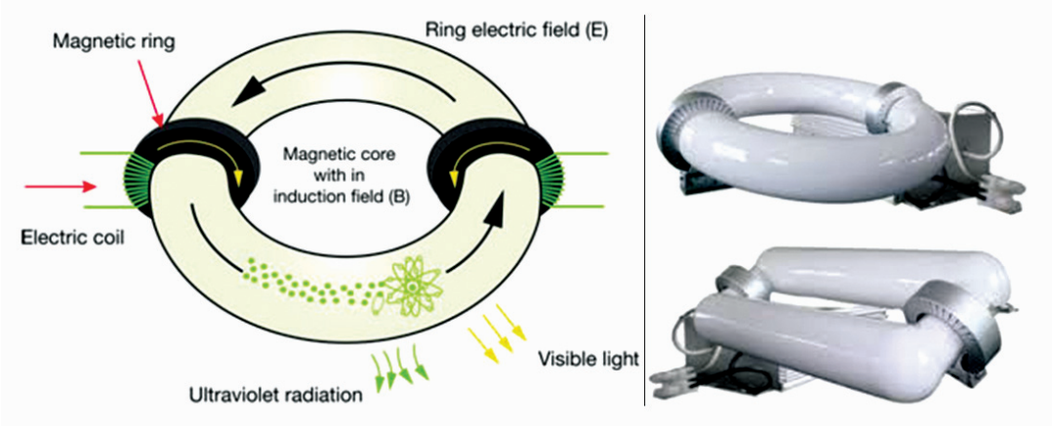


Figure 8.12 Representation of Induction Lamp

As the electrons accelerate inside the bulb, they collide with mercury atoms and produce ultraviolet (UV) light radiation. The UV light then causes the special phosphor coating inside the glass to react in a way that produces fluorescent light within the visible spectrum. The light produced by Induction Lighting (Figure 8.12) achieves good Color Rendering Index (CRI), with a Correlated Color Temperature.

Advantages of Induction lamps is as follows long burning hours, very less maintenance required, instant on/ instant re-strike and energy efficient lighting.

Lamp features: The Table 8.1 shows the lamp features of different lamps.

Table 8.1 Luminous Performance Characteristics of Commonly Used Luminaries

Type of Lamp	Lumens / Watt		Colour Rendering Index	Typical Application	Typical Life (hours)
	Range	Avg.			
Incandescent	8-18	14	Excellent (100)	Homes, restaurants, general lighting, emergency lighting	1000
Fluorescent lamps	46-60	50	Good w.r.t. coating (67-77)	Offices, shops, hospitals, homes	5000
Compact fluorescent lamps (CFL)	40-70	60	Very good (85)	Hotels, shops, homes, offices	8000-10000
High pressure mercury (HPMV)	44-57	50	Fair (45)	General lighting in factories, garages, car parking, flood lighting	5000
Halogen lamps	18-24	20	Excellent (100)	Display, flood lighting, stadium exhibition grounds, construction areas	2000-4000
High pressure sodium (HPSV) SON	67-121	90	Fair (22)	General lighting in factories, ware houses, street lighting	6000-12000
Low pressure sodium (LPSV) SOX	101-175	150	Poor (10)	Roadways, tunnels, canals, street lighting	6000-12000
Metal halide lamps	75-125	100	Good (70)	Industrial bays, spot lighting, flood lighting, retail stores	8000
LED lamps	50-130	90	Very good (80)	Office, industry, outdoor, retail, hospitality, etc	30,000-60,000
Induction Lamps	65-90	75	Very good (80)	General lighting, factories, warehouse, street lighting, flood lighting, etc	60,000-1,00,000

8.4 Recommended Illuminance Levels for Various Tasks / Activities / Locations

Recommendations on Illuminance

Scale of Illuminance: The minimum illuminance for all non-working interiors, has been mentioned as 20 Lux (as per IS 3646). A factor of approximately 1.5 represents the smallest significant difference in subjective effect of illuminance. Therefore, the following scale of illuminances is recommended.

20–30–50–75–100–150–200–300–500–750–1000–1500–2000, … Lux

Illuminance ranges: Because circumstances may be significantly different for different interiors used for the same application or for different conditions for the same kind of activity, a range of illuminances is recommended for each type of interior or activity intended of a single value of illuminance. Each range consists of three successive steps of the recommended scale of illuminances. For working interiors the middle value (R) of each range represents the recommended service illuminance that would be used unless one or more of the factors mentioned below apply.

The higher value (H) of the range should be used at exceptional cases where low reflectance or contrasts are present in the task, errors are costly to rectify, visual work is critical, accuracy or higher productivity is of great importance and the visual capacity of the worker makes it necessary.

Similarly, lower value (L) of the range may be used when reflectances or contrasts are unusually high, speed and accuracy is not important and the task is executed only occasionally.

Recommended Illumination

The following Table 8.2 gives the recommended illuminance range for different tasks and activities for chemical sector. The values are related to the visual requirements of the task, to user's satisfaction, to practical experience and to the need for cost effective use of energy (Source IS 3646 (Part I): 1992). For recommended illumination in other sectors, reader may refer *Illuminating Engineers Society Recommendations Handbook*.

Table 8.2 Recommended illuminance range for different tasks and activities for chemical sector

Petroleum, Chemical and Petrochemical works	
Exterior walkways, platforms, stairs and ladders	30-50-100
Exterior pump and valve areas	50-100-150
Pump and compressor houses	100-150-200
Process plant with remote control	30-50-100
Process plant requiring occasional manual intervention	50-100-150
Permanently occupied work stations in process plant	150-200-300
Control rooms for process plant	200-300-500

Pharmaceuticals Manufacturer and Fine chemicals manufacturer	
Pharmaceutical manufacturer	
Grinding, granulating, mixing, drying, tableting, sterilising, washing, preparation of solutions, filling, capping, wrapping, hardening	300-500-750
Fine chemical manufacturers	
Exterior walkways, platforms, stairs and ladders	30-50-100
Process plant	50-100-150
Fine chemical finishing	300-500-750
Inspection	300-500-750
Soap manufacture	
General area	200-300-500
Automatic processes	100-200-300
Control panels	200-300-500
Machines	200-300-500
Paint works	
General	200-300-500
Automatic processes	150-200-300
Control panels	200-300-500
Special batch mixing	500-750-1000
Colour matching	750-100-1500

8.5 Methods of Calculating Illuminance - Lighting Design for Interiors

In order to design a luminaire layout that best meets the illuminance and uniformity requirements of the job, two types of information are generally needed: average illuminance level and illuminance level at a given point. Calculation of illuminance at specific points is often done to help the designer evaluate the lighting uniformity, especially when using luminaires where maximum spacing recommendations are not supplied, or where task lighting levels must be checked against ambient level.

If average levels are to be calculated, two methods can be applied:

1. For indoor lighting situations, the Zonal Cavity Method is used with data from a coefficient of utilization table.
2. For outdoor lighting applications, a coefficient of utilization curve is provided, the CU is read directly from the curve and the standard lumen formula is used.

Zonal Cavity Method for Indoor Lighting Calculations

The Zonal Cavity Method (sometimes called the Lumen Method) is the currently accepted method for calculating average illuminance levels for indoor areas, unless the light distribution is radically asymmetric. It is an accurate hand method for indoor applications because it takes into consideration the effect that inter-reflectance has on the level of illuminance.

Although it takes into account several variables, the basic premise that foot-candles are equal to luminous flux over an area is not violated.

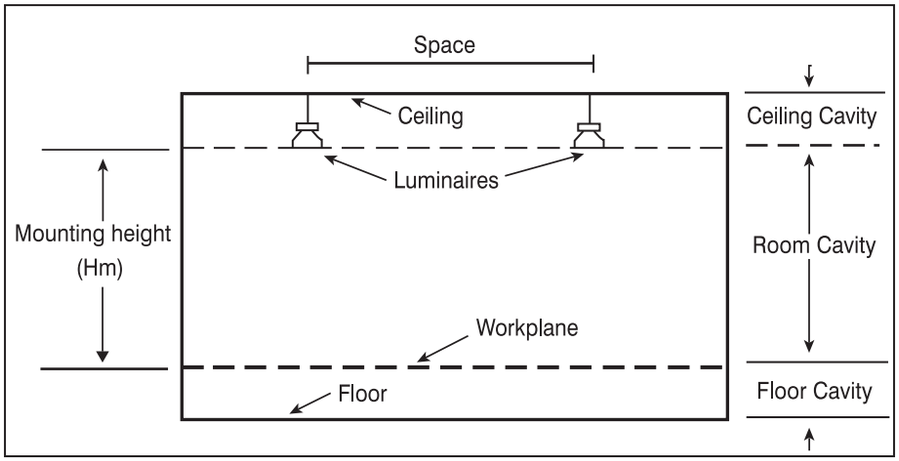


Figure 8.13 Typical Room Space

The basis of the Zonal Cavity Method is that a room is made up of three spaces or cavities. The space between the ceiling and the fixtures, if they are suspended, is defined as the "ceiling cavity"; the space between the work plane and the floor, the "floor cavity"; and the space between the fixtures and the work plane, the "room cavity."

Example:

The step by step process of lighting design is illustrated below with the help of an example. The Figure 8.13 shows the parameters of a typical space.

Step-1: Decide the required illuminance on work plane, the type of lamp and luminaire

A preliminary assessment must be made of the type of lighting required, a decision most often made as a function of both aesthetics and economics. For normal office work, illuminance of 200 lux is desired.

For an air conditioned office space under consideration, we choose 36 W fluorescent tube lights with twin tube fittings. The luminaire is porcelain-enameled suitable for the above lamp. It is necessary to procure utilisation factor tables for this luminaire from the manufacturer for further calculations.

Step-2: Collect the room data in the format given below

Room dimensions	Length	L1	10	m
	Width	L2	10	m
	Floor area	L3	100	m^2
	Ceiling height	L4	3.0	m
Surface reflectance	Ceiling	L5	0.7	p.u
	Wall	L6	0.5	p.u
	Floor	L7	0.2	p.u
Work plane height from floor		L8	0.9	m
Luminaire height from floor		L9	2.9	m

Typical reflectance values for using in L5, L6, L7 are:

	Ceiling	Walls	Floor
Air conditioned office	0.7	0.5	0.2
Light industrial	0.5	0.3	0.1
Heavy industrial	0.3	0.2	0.1

Step-3: Calculate room index

$$\textbf{Room Index} = \frac{\textbf{Length} \times \textbf{Width}}{\textbf{Mounting Height} \times (\textbf{Length} + \textbf{Width})}$$

$$= \frac{L1 \times L2}{(L9 - L8) \times (L1 + L2)} = \frac{10 \times 10}{2 \times (10 + 10)}$$

$$= 2.5$$

Step 4: Calculate the utilisation factor

Utilisation factor is defined as the percent of rated bare-lamp lumens that exit the luminaire and reach the work plane. It accounts for light directly from the luminaire as well as light reflected off the room surfaces. Manufacturers will supply each luminaire with its own CU table derived from a photometric test report.

Using tables available from manufacturers, it is possible to determine the utilisation factor for different light fittings if the reflectance of both the walls and ceiling is known, the room index has been determined and the type of luminaire is known. For twin tube fixture, utilisation factor is 0.66, corresponding to room index of 2.5.

Step-5: To calculate the number of fittings required, the following formula is used

$$\mathbf{N} = \frac{\mathbf{E} \times \mathbf{A}}{\mathbf{F} \times \mathbf{UF} \times \mathbf{LLF}}$$

Where, N = Number of Fittings

E = Lux Level Required on Working Plane

A = Area of Room (L x W)

F = Total Flux (Lumens) from all the Lamps in one Fitting

UF = Utilisation Factor from the Table for the Fitting to be Used

LLF = Light Loss Factor. This takes account of the depreciation over time of lamp output and dirt accumulation on the fitting and walls of the building.

Light output = 3050 Lumens (Single Lamp)

LLF = Lamp lumen MF x Luminaire MF x Room surface MF

Where, MF = Maintenance Factor

Typical LLF values

Air conditioned office	0.8
Clean industrial	0.7
Dirty industrial	0.6

$$N = \frac{200 \times 100}{2 \times 3050 \times 0.66 \times 0.8} = 6.2$$

So, 6 Numbers of Twin Tube Fixtures are required. Total number of 36 W lamp is 12.

Step 6: Space the luminaires to achieve desired uniformity

Every luminaire will have a recommended space to height ratio. In earlier design methodologies, the uniformity ratio, which is the ratio of minimum illuminance to average illuminance, was kept at 0.8 and suitable space to height ratio is specified to achieve the uniformity. In modern designs incorporating energy efficiency and task lighting, the emerging concept is to provide a uniformity of 1/3 to 1/10 depending on the tasks.

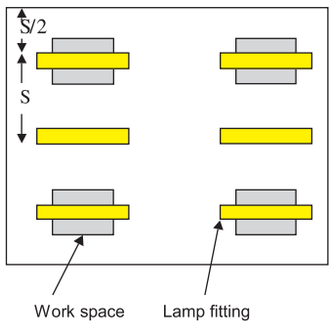


Figure 8.14 Arrangement of Fittings

Recommended value for the above luminaire is 1.5. If the actual ratio is more than the recommended values, the uniformity of lighting will be less.

For a sample of arrangement of fittings, refer Figure 8.14. The luminaire closer to a wall should be one half of spacing or less.

Luminaire Spacing

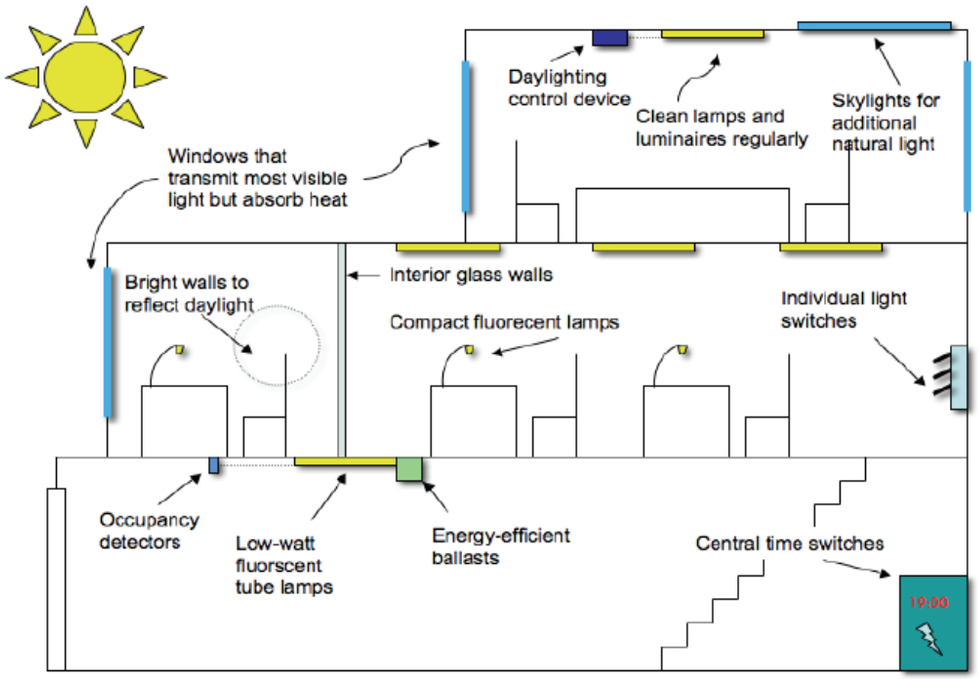
Spacing between luminaires	= 10/3	= 3.33 m
Mounting height (L9-L8)		= 2.0 m
Space to height ratio (SHR)	= 3.33/2.0	= 1.66

This is close to the limits specified and hence accepted.

It is better to choose luminaires with larger SHR. This can reduce the number of fittings and connected lighting load.

8.6 General Energy Saving Opportunities

Changing the light bulbs is not the only way to improve the use of lighting. Below are some examples of many other options available:



a) Use natural day lighting

The utility of using natural day lighting instead of electric lighting during the day is well known, but is being increasingly ignored especially in modern air-conditioned office spaces and commercial establishments like hotels, shopping plazas etc. Industrial plants generally use daylight in some fashion, but improperly designed day lighting systems can result in complaints from personnel or supplementary use of electric lights during daytime.

Some of the methods to incorporate day lighting are:

i. North lighting by use if single-pitched truss of the saw-tooth type is a common industrial practice; this design is suitable for latitudes north of 23 i.e. in North India. In South India, north lighting may not be appropriate unless diffusing glasses are used to cut out the direct sunlight.

ii. Innovative designs are possible which eliminates the glare of daylight and blend well with the interiors. Glass strips, running continuously across the breadth of the roof at regular intervals, can provide good, uniform lighting on industrial shop floors and storage bays.

iii. A good design incorporating sky lights with FRP material along with transparent or translucent false ceiling can provide good glare-free lighting; the false ceiling will also cut out the heat that comes with natural light.

iv. Use of atrium with FRP dome in the basic architecture can eliminate the use of electric lights in passages of tall buildings.

v. Natural Light from windows should also be used. However, it should be well designed to avoid glare. Light shelves can be used to provide natural light without glare.

vi. Mounting Solar tube on the roof, with the help of advanced optics and special duct work to direct sunlight deep into the buildings and spreading out over large internal spaces providing heat and glare free daylighting for 8-10 hrs in a day

b) De-lamping to reduce excess lighting

De-lamping is an effective method to reduce lighting energy consumption. In some industries, reducing the mounting height of lamps, providing efficient luminaires and then de-lamping has ensured that the illuminance is hardly affected. De-lamping at empty spaces where active work is not being performed is also a useful concept.

c) Task lighting

Task Lighting implies providing the required good illuminance only in the actual small area where the task is being performed, while the general illuminance of the shop floor or office is kept at a lower level; e.g. Machine mounted lamps or table lamps. Energy saving takes place because good task lighting can be achieved with low wattage lamps. The concept of task lighting if sensibly implemented, can reduce the no of general lighting fixtures, reduce the wattage of lamps, save considerable energy and provide better illuminance and also provide aesthetically pleasing ambience.

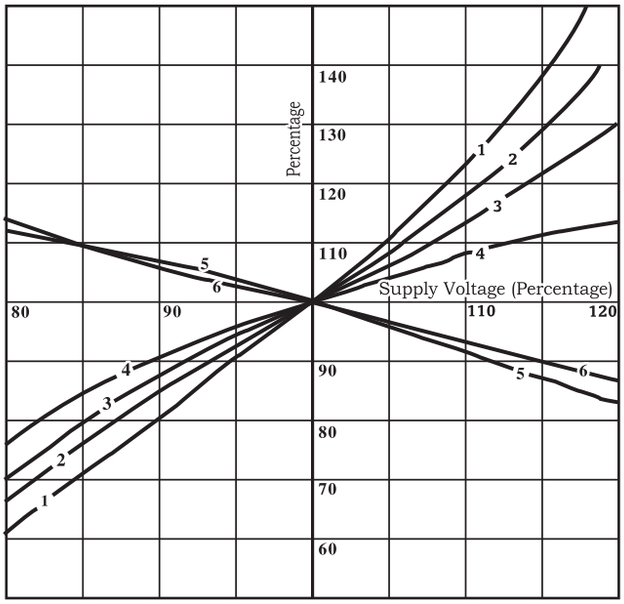
d) Selection of high efficiency lamps and luminaries

The details of common types of lamps are summarised in Table 8.1 above. It is possible to identify energy saving potential for lamps by replacing with more efficient types. The following examples of lamp replacements are common. There may be some limitations if colour rendering is an important factor. It may be noted that, in most cases, the luminaires and the control gear would also have to be changed. The savings are large if the lighting scheme is redesigned with higher efficacy lamps and luminaires.

e) Reduction of lighting feeder voltage

Figure 8.15 shows the effect of variation of voltage on light output and power consumption for fluorescent tube lights. Similar variations are observed on other gas discharge lamps like mercury vapour lamps, metal halide lamps and sodium vapour lamps (Table 8.3 summarises the effects). Hence reduction in lighting feeder voltage can save energy, provided the drop in light output is acceptable.

In many areas, night time grid voltages are higher than normal; hence reduction in voltage can save energy and also provide the rated light output. Some manufacturers are supplying reactors and transformers as standard products. A large number of industries have used these devices and have reported saving to the tune of 5% to 15%. Industries having a problem of higher night time voltage can get an additional benefit of reduced premature lamp failures.



1. Lamp Current 2. Circuit Power 3. Lamp Power 4. Lamp Output
5. Lamp Voltage 6. Lamp Efficacy

Figure 8.15 Effect of Voltage Variation on Fluorescent Tube light Parameters

Table 8.3 Variation in Light Output and Power Consumption

Particulars	10% lower voltage	10% higher voltage
Fluorescent lamps		
Light output	Decreases by 9 %	Increases by 8 %
Power input	Decreases by 15 %	Increases by 8 1%
HPMV lamps		
Light output	Decreases by 20 %	Increases by 20 %
Power input	Decreases by 16 %	Increases by 17 %
Mercury Blended lamps		
Light output	Decreases by 24 %	Increases by 30 %
Power input	Decreases by 20 %	Increases by 20 %
Metal Halide lamps		

f) Electronic ballasts

Conventional electromagnetic ballasts (chokes) are used to provide higher voltage to start the tube light and subsequently limit the current during normal operation. *Electronic ballasts* are oscillators that convert the supply frequency to about 20,000 Hz to 30,000 Hz. The basic functions of electronic ballast are:

- To ignite the lamp
- To stabilize the gas discharge
- To supply the power to the lamp

The losses in electronic ballasts for tube lights are only about 1 Watt, in place of 10 to 15 Watts in standard electromagnetic chokes.

The additional advantage is that the efficacy of tube lights improves at higher frequencies, resulting in additional savings if the ballast is optimised to provide the same light output as with the conventional choke. Hence a saving of about 15 to 20 Watts per tube light can be achieved by use of electronic ballasts. With electronic ballast, the starter is eliminated and the tube light lights up instantly without flickering.

g) Lighting controllers

Automatic control for switching off unnecessary lights can lead to good energy savings. This includes dimmers, motion & occupancy sensors, photosensors and timers.

h) Lighting maintenance

Maintenance is vital to lighting efficiency. Light levels decrease over time because of aging lamps and dirt on fixtures, lamps and room surfaces. Together, these factors can reduce total illumination by 50 percent or more, while lights continue drawing full power. The basic maintenance includes cleaning of lamps and fixtures, cleaning and repainting interiors, relamping etc

8.7 Energy Efficient Lighting Controls

Occupancy Sensors

Occupancy-linked control can be achieved using infra-red, acoustic, ultrasonic or microwave sensors, which detect either movement or noise in room spaces. These sensors switch lighting on when occupancy is detected, and off again after a set time period, when no occupancy movement detected. They are designed to override manual switches and to prevent a situation where lighting is left on in unoccupied spaces. With this type of system it is important to incorporate a built-in time delay, since occupants often remain still or quiet for short periods and do not appreciate being plunged into darkness if not constantly moving around.

Timed Based Control

Timed-turnoff switches are the least expensive type of automatic lighting control. In some cases, their low cost and ease of installation makes it desirable to use them where more efficient controls would be too expensive.

Types and features

The oldest and most common type of timed-turnoff switch is the “dial timer,” a spring-wound mechanical timer that is set by twisting the knob to the desired time. Typical units of this type are vulnerable to damage because the shaft is weak and the knob is not securely attached to the shaft. Some spring-wound units make an annoying ticking sound as they operate. Newer types of timed-turnoff switches are completely electronic and silent. Electronic switches can be made much more rugged than the

spring-wound dial timer. These units typically have a spring-loaded toggle switch that turns on the circuit for a preset time interval. Some electronic models provide a choice of time intervals, which you select by adjusting a knob located behind the faceplate. Most models allow occupants to turn off the lights manually. Some models allow occupants to keep the lights on, overriding the timer. Timed-turnoff switches are available with a wide range of time spans. The choice of time span is a compromise. Shorter time spans waste less energy but increase the probability that the lights will turn off while someone is in the space. Dial timers allow the occupant to set the time span, but this is not likely to be done with a view toward optimising efficiency. For most applications, the best choice is an electronic unit that allows the engineering staff to set a fixed time interval behind the cover plate.

Daylight Linked Control

Photoelectric cells can be used either simply to switch lighting on and off, or for dimming. They may be mounted either externally or internally. It is however important to incorporate time delays into the control system to avoid repeated rapid switching caused, for example, by fast moving clouds. By using an internally mounted photoelectric dimming control system, it is possible to ensure that the sum of daylight and electric lighting always reaches the design level by sensing the total light in the controlled area and adjusting the output of the electric lighting accordingly. If daylight alone is able to meet the design requirements, then the electric lighting can be turned off. The energy saving potential of dimming control is greater than a simple photoelectric switching system. Dimming control is also more likely to be acceptable to room occupants.

Localized Switching

Localized switching should be used in applications which contain large spaces. Local switches give individual occupants control over their visual environment and also facilitate energy savings. By using localized switching it is possible to turn off artificial lighting in specific areas, while still operating it in other areas where it is required, a situation which is impossible if the lighting for an entire space is controlled from a single switch.

Street Lighting Systems and Controls

Street lighting /Public lighting is one of the major electrical loads in municipal areas. Number of street lights used in a Municipal area varies from 20000 – 50000 in numbers depending on the kilometers of road illuminated within the municipal limits. Typical electrical load of municipal lighting system varies 2MW to 7 MW. The type of lamps used in Municipal area includes Fluorescent Tube light/ Mercury Vapor Lamps/ Sodium Vapor Lamps and Metal Halide Lamps. High Mast towers are also used at strategic junctions in the Municipal area. LEDs are also used for traffic signaling purpose in municipal areas.

Following controls are adopted to reduce energy consumption in street lighting system:

1. Timer control (Switch ON/OFF as per set timing)
2. Day light control(Based on illumination level)
3. Selective switching/Alternate switching of street lights low traffic density areas (after midnight).

4. Switching control based on lux levels. (after midnight)
5. Installations of Voltage controllers to be operated after midnight.
6. Installation of PLC controlled Lighting panels for effective control and monitoring.

8.8 Standards and Labeling Programs for FTL Lamps

Considering the large number of fluorescent lamps (FTL) in usage, BEE has included FTL under Standard and Labeling Programme (S&L). The S&L Programme covers 4 feet tubular fluorescent lamps (101mm) for wattages up to 40W. The S&L programme includes 6500K colour temperature for halo-phosphates and 6500K, 4000K & 2700K for tri-phosphate category. The star rating scheme for FTL is given in Table 8.4.

Table 8.4 Star Rating scheme for FTL (101 mm)

	1 Star	2 Star	3 Star	4 Star	5 Star
Lumens per Watt at 0100 hrs of use	<61	>=61 & <67	>=67 & <86	>=86 & <92	>=92
Lumens per Watt at 2000 hrs of use	<52	>=52 & <57	>=57 & <77	>=77 & <83	>=83
Lumens per Watt at 3500 hrs of use	<49	>=49 & <54	>=54 & <73	>=73 & <78	>=78

8.9 Lighting Case Study

Replacement of existing T12 Fluorescent lamps in street lighting system with LED lamps

Existing: Fluorescent lamp (T12) fixture of 40 numbers is connected to the entire campus for security purpose. All the lights remain in operation for around 12 hours at night (6 p.m. to 6 am) every day throughout the year. All the light fixtures are equipped with electromagnetic ballast which consumes around 12 to 14 watt of additional power while in operation. Hence the power consumption of a single fluorescent light fixture considering minimum ballast loss is 40+12=52 watts. The total light output of all the fluorescent light fixtures is around 2400 lumen.

Proposed: It was proposed to replace existing lamps with high efficient LED lamps of 18 W with a luminous efficacy of around 120-140 lm/w. The total luminous output of these lamps is around 2340 lumen.

Calculation

Existing Fittings			
No of FTL-T12 lamps installed	=	40	No's
Wattage Consumed			
40 No's of (40W+12W (Ballast)=52 W)	=	2.1	kW
Average operating hours per day	=	12	hrs/day

Total energy consumed by operating the lights	=	25	kWh/day
Annual energy consumption by operating the lights (365 days/year)	=	9125	kWh/annum
Proposed option			
Replacement of all 40 no's of 40 Watts T12-FTL with 18 Watts LED lamps			
Total energy consumed by operating with LED	=	0.72	kW
Average operating hours per day	=	12	hrs/day
Total kWh/day consumed by operating the lights	=	9	kWh/day
Annual energy consumption by operating the lights (365 days/year)	=	3285	kWh/annum
Savings			
Total energy reduction per annum	=	5840	kWh/annum
Annual monetary savings (@Rs. 5/unit)	**=**	***29,200***	***Rs./annum***
Investment @ **Rs.2500/ Lamp**	=	1.0	Lakhs
Simple payback period	=	3.4	Years

QUESTIONS	
Objective Type Questions	
1.	The unit of one lux is a) 1000 lumen per square feet b) 10 lumen per square meter c) 1 lumen per square meter d) 1 lumen per square feet
2.	Luminous efficacy of a lamp is given by a) Lux/Watt b) lumens/Watt c) Watt/Lux d) Watt/lumens
3.	A fluorescent tube light fitted with an electronic choke will a) operate at 50 Hz b) not need a starter c) operate at 0.5 power factor d) none of the above
4.	A device that distributes and filters the light emitted from one or more lamps is a) control gear b) lamp c) luminaire d) starter
5.	The T2,T5,T8 and T12 fluorescent lamps are categorized based on a) diameter of the tube b) length of the tube c) both diameter and length d) none of the above
6.	Which of the following options reduces the lighting consumption in the wide spread plant? a) replacing 150 W HPSV lamps with 250 W HPMV lamps b) maintaining 260 V for the lighting circuit with 220 V rated lamps c) installing separate lighting transformer and maintaining optimum voltage d) none of the above
7.	Which of the following lamp has the highest efficacy? a) Metal halide b) halogen lamps c) HPMV d) HPSV
8.	A CFL is an energy efficient replacement for a) FTL b) GLS c) HPMV d) HPSV
9.	Which method uses infrared, acoustic, ultrasonic or microwave sensors for lighting control? a) time-based control b) daylight-linked control c) occupancy-linked control d) localized switching
10.	Which of the following is the best definition of illuminance? a) Time rate of flow of light energy b) Luminous flux incident on an object per unit area c) Flux density emitted from an object without regard for direction d) Flux density emitted from an object in a given direction

Short Type Questions	
S-1	What is the function of an electronic choke in a lighting system?
S-2	List five energy saving measures in lighting system
S-3	Define the terms Lux, CRI, Luminous efficacy?
S-4	In one of the municipal area each 250 watt HPSV lamp is proposed to be replaced by 2 numbers of 28 watts CFL lamp. What is the annual saving if the lamp is operated for 3000 hour in a year? Do you recommend / reject such energy saving measure. Give reason.
S-5	The illuminance is 17 lm/m² from a lamp at 2 meter distance. What will be the illuminance at 1.5 m distance?
Long Type Questions	
L-1	Explain the principle of operation and features of LED lamps, Incandescent lamp, Fluorescent lamp and metal halide lamps?
L-2	A municipal corporation has 30000 numbers of 250 Watts HPMV lamps. The annual operating hours of these lamps are 3600 hours. Calculate the annual energy consumption and the annual cost if the electricity tariff is Rs 4.75/kWh. Estimate the demand cost if the present demand cost is Rs 150 / KVA. Assume 0.7 PF for the lighting load. Discuss the various energy savings opportunity in the municipal lighting system with investment and simple payback for minimum 3 savings options.

REFERENCES

1. Best Practices Lighting Manual, BEE Code, 2006
2. NPC Energy Audit Reports
3. Lighting Handbook

Chapter 9

Diesel / Natural Gas Power Generating System

Learning Objectives

In this chapter you will learn about

- Basics of internal combustion engines
- Diesel/Natural gas power generating system
- Factors affecting performance
- Waste heat recovery in Diesel / Natural gas power generating system
- Performance assessment methodology
- Energy saving measures

9. DIESEL / NATURAL GAS POWER GENERATING SYSTEM

Factors affecting selection, Waste heat recovery, Energy performance assessment of DG sets, Diesel conservation opportunities.

9.1 Introduction

Reciprocating engines produce electricity using a combustible fuel and generator. In addition to producing electricity, useful heat can be recovered from the exhaust gas using a heat recovery steam generator (HRSG), or heat recovery system for hot water (Figure 9.1). Heat can also be recovered from the lubricating oil cooler, the jacket water cooler and/or the charge air cooler, and this recovered "waste" heat can be provided to a heating load. In this case, the reciprocating engine power plant would be operating in a Combined Heat & Power (CHP) or cogeneration mode. The energy performance of a reciprocating engine is influenced by a number of factors such as the type of fuel, the reciprocating engine power capacity, minimum capacity, availability, heat rate and heat recovery efficiency.

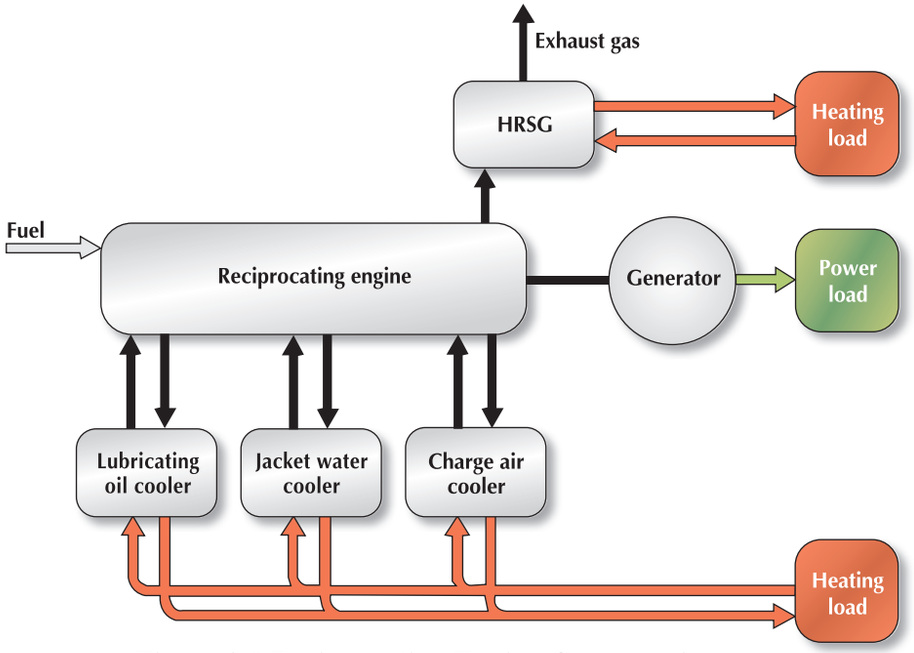


Figure 9.1 Reciprocating Engine Cogeneration

There are two basic types of reciprocating engines - spark ignition and compression ignition. Spark ignition engines use a spark (across a spark plug) to ignite a compressed fuel-air mixture. Typical fuels for such engines are gasoline, natural gas and sewage and landfill gas. Compression ignition engines compress air to a high pressure, heating the air to the ignition temperature of the fuel, which then is injected. The high compression ratio used for compression ignition engines results in a higher efficiency than is possible with spark ignition engines. Diesel/heavy fuel oil is normally used in compression ignition engines, although some are dual-fueled (natural gas is compressed with the combustion air and diesel oil is injected at the top of the compression stroke to initiate combustion).

Diesel Engine Cycle

Diesel engine is the prime mover, which drives an alternator to produce electrical energy. In the Diesel engine, air is drawn into the cylinder and is compressed to a high ratio (14:1 to 25:1). During this compression, the air is heated to a temperature of 700–900^{0}C. A metered quantity of diesel fuel is then injected into the cylinder, which ignites spontaneously because of the high temperature. Hence, the diesel engine is also known as compression ignition (CI) engine.

DG set can be classified according to cycle type as: two stroke and four stroke. However, the bulk of IC engines use the four stroke cycle. Let us look at the principle of operation of the four-stroke diesel engine.

The 4 stroke operations in a diesel engine (Figure 9.2) are: induction stroke, compression stroke, ignition and power stroke and exhaust stroke.

1st : Induction stroke - while the inlet valve is open, the descending piston draws in fresh air.

2nd : Compression stroke - while the valves are closed, the air is compressed to a pressure of up to 25 bar.

3rd : Ignition and power stroke - fuel is injected, while the valves are closed (fuel injection actually starts at the end of the previous stroke), the fuel ignites spontaneously and the piston is forced downwards by the combustion gases.

4th : Exhaust stroke - the exhaust valve is open and the rising piston discharges the spent gases from the cylinder.

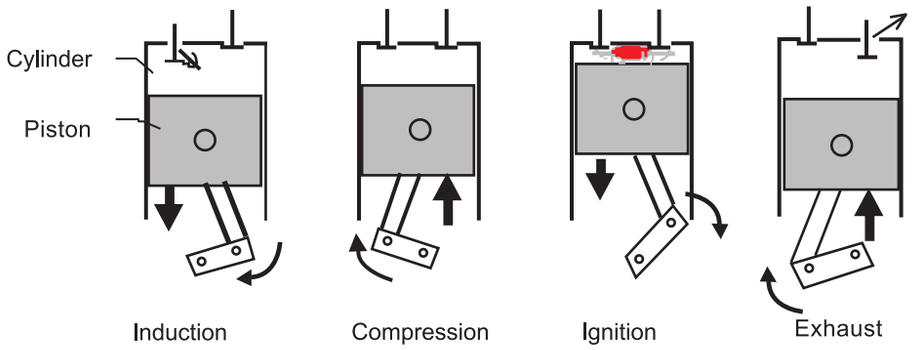


Figure 9.2 Schematic Diagram of Four-Stroke Diesel Engine

Since power is developed during only one stroke, the single cylinder four-stroke engine has a low degree of uniformity. Smoother running is obtained with multi cylinder engines because the cranks are staggered in relation to one another on the crankshaft. There are many variations of engine configuration, for example. 4 or 6 cylinder, in-line, horizontally opposed, vee or radial configurations.

Gas Engines

A typical spark-ignited lean-burn engine is depicted in Figure 9.3. In this process, the gas is mixed with air before the inlet valves. During the intake period, gas is also fed into a small prechamber, where

the gas mixture is rich compared to the gas in the cylinder. At the end of the compression phase the gas/air mixture in the prechamber is ignited by a spark plug. The flames from the nozzle of the prechamber ignite the gas/air mixture in the whole cylinder. Combustion is fast. After the working phase the cylinder is emptied of exhaust and the process starts again. Reciprocating engines with modern lean-burn technology reach close to 45% electrical efficiency.

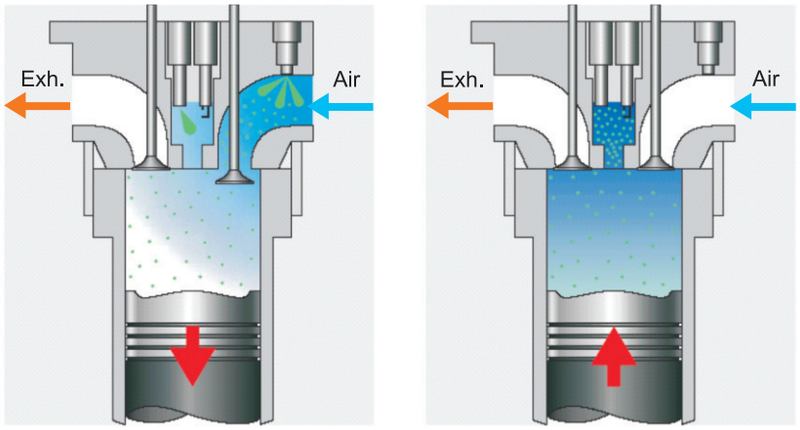


Figure 9.3 Natural Gas Engine

DG Set as a System

A diesel generating set (Figure 9.4) should be considered as a system since its successful operation depends on the well-matched performance of the components, namely:

a) The diesel engine and its accessories.
b) The AC Generator.
c) The control systems and switchgear.
d) The foundation and power house civil works.
e) The connected load with its own components like heating, motor drives, lighting etc.

It is necessary to select the components with highest efficiency and operate them at their optimum efficiency levels to conserve energy in this system.

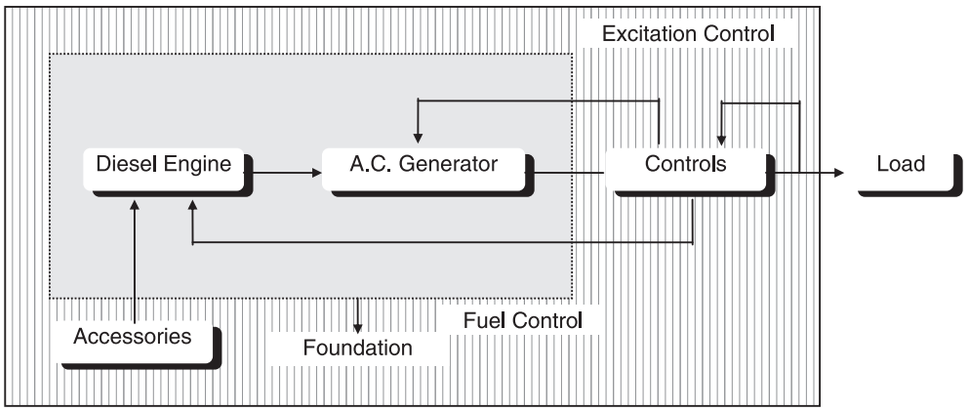


Figure 9.4 DG Set System

Selection Considerations

To make a decision on the type of engine, which is most suitable for a specific application, several factors need to be considered. The two most important factors are: power and speed of the engine.

The power requirement is determined by the maximum load. The engine power rating should be 10-20 % more than the power demand by the end use. This prevents overloading the machine by absorbing extra load during starting of motors or switching of some types of lighting systems or when wear and tear on the equipment pushes up its power consumption.

Speed is measured at the output shaft and given in revolutions per minute (RPM). An engine will operate over a range of speeds, with diesel engines typically running at lower speeds (1300 - 3000 RPM). There will be an optimum speed at which fuel efficiency will be greatest. Engines should be run as closely as possible to their rated speed to avoid poor efficiency and to prevent build up of engine deposits due to incomplete combustion - which will lead to higher maintenance and running costs. To determine the speed requirement of an engine, one has to again look at the requirement of the load.

For some applications, the speed of the engine is not critical, but for other applications such as a generator, it is important to get a good speed match. If a good match can be obtained, direct coupling of engine and generator is possible; if not, then some form of gearing will be necessary - a gearbox or belt system, which will add to the cost and reduce the efficiency.

There are various other factors that have to be considered, when choosing an engine for a given application. These include the following: cooling system, abnormal environmental conditions (dust, dirt, etc.), fuel quality, speed governing (fixed or variable speed), poor maintenance, control system, starting equipment, drive type, ambient temperature, altitude, humidity, etc.

Suppliers or manufacturers literature will specify the required information when purchasing an engine. The efficiency of an engine depends on various factors, for example, load factor (percentage of full load), engine size, and engine type.

Diesel Generator Captive Power Plants

Diesel engine power plants are most frequently used in small power (captive non-utility) systems. The main reason for their extensive use is the higher efficiency of the diesel engines compared with gas turbines and small steam turbines in the output range considered. In applications requiring low captive power, without much requirement of process steam, the ideal method of power generation would be by installing diesel generator plants. The fuels burnt in diesel engines range from light distillates to residual fuel oils. Most frequently used diesel engine sizes are between the range 4 to 15 MW. For continuous operation, low speed diesel engine is more cost-effective than high speed diesel engine.

Advantages of adopting Diesel Power Plants are:

- Low installation cost
- Short delivery periods and installation period
- Higher efficiency (as high as 43 -45 %)
- More efficient plant performance under part loads

- Suitable for different type of fuels such as low sulphur heavy stock and heavy fuel oil in case of large capacities.
- Minimum cooling water requirements,
- Adopted with air cooled heat exchanger in areas where water is not available
- Short start up time

A brief comparison of different types of captive power plants (combined gas turbine and steam turbine, conventional steam plant and diesel engine power plant) is given in Table 9.1. It can be seen from the Table that captive diesel plant wins over the other two in terms of thermal efficiency, capital cost, space requirements, auxiliary power consumption, plant load factor etc.

Table 9.1 Comparison of Different Types of Captive Power Plant

Description	Units	Combined GT & ST	Conventional Steam Plant	Diesel Engine Power Plants
Thermal Efficiency	%	40 - 46	33 - 36	43 - 45
Initial Investment of Installed Capacity	Rs./kW	8,500 - 10,000	15,000 -18,000	7,500 - 9,000
Space requirement		125 % (Approx.)	250 % (Approx.)	100 % (Approx.)
Construction time	Months	24 - 30	42 - 48	12 - 15
Project period	Months	30 - 36	52 - 60	12
Auxiliary Power Consumption	%	2 – 4	8 - 10	1.3 – 2.1
Plant Load Factor	kWh/kW	6000 - 7000	5000 - 6000	7200 - 7500
Start up time from cold	Minutes	About 10	120 - 180	15 - 20

Diesel Engine Power Plant Developments

The diesel engine developments have been steady and impressive. The specific fuel consumption has come down from a value of 220 g/kWh in the 1970's to a value of around 160 g/kWh in present times.

Slow speed diesel engine, with its flat fuel consumption curve over a wide load range (50%-100%), compares very favorably over other prime movers such as medium speed diesel engine, steam turbines and gas turbines.

With the arrival of modern, high efficiency turbochargers, it is possible to use an exhaust gas driven turbine generator (Figure 9.5) to further increase the engine rated output. The net result would be lower fuel consumption per kWh and further increase in overall thermal efficiency.

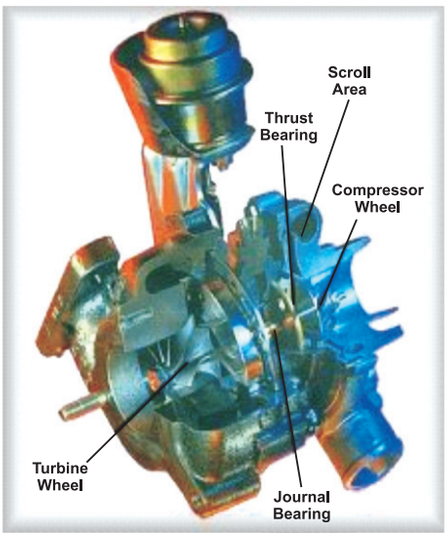


Figure 9.5 Turbocharger

The diesel engine is able to burn the poorest quality fuel oils, unlike gas turbine, which is able to do so with only costly fuel treatment equipment.

Slow speed dual fuel engines are now available using high-pressure gas injection, which gives the same thermal efficiency and power output as a regular fuel oil engine.

9.2 Selection and Installation Factors

Sizing of a Genset:

a) If the DG set is required for 100% standby, then the entire connected load in HP / kVA should be added. After finding out the diversity factor, the correct capacity of a DG set can be found out.

Example :

Connected Load	=	650 kW
Diversity Factor (Demand / Connected load)	=	0.54
Max. Demand	=	650 x 0.54 = 350 kW
% Loading	=	70
Set rating	=	350/0.7 = 500 kW
At 0.8 PF, rating	=	625 kVA

b) For an existing installation, record the current, voltage and power factors (kWh / kVAh) reading at the main bus-bar of the system at every half-an-hour interval for a period of 2-3 days and during this period the factory should be having its normal operations. The non-essential loads should be switched off to find the realistic current taken for running essential equipment. This will give a fair idea about the current taken from which the rating of the set can be calculated.

For existing installation:

kVA	=	$\sqrt{3}$ V I
kVA Rating	=	kVA / Load Factor
where Load factor	=	Average kVA / Maximum kVA

c) For a new installation, an approximate method of estimating the capacity of a DG set is to add full load currents of all the proposed loads to be run in DG set. Then, applying a diversity factor depending on the industry, process involved and guidelines obtained from other similar units, correct capacity can be arrived at.

High Speed Engine or Slow/Medium Speed Engine

The normal accepted definition of high speed engine is 1500 rpm. The high speed sets have been developed in India, whereas the slow speed engines of higher capacities are often imported. The other features and comparison between high and medium / slow speed engines are mentioned in Table 9.2 below:

Table 9.2 Comparison of High and Slow Speed Engine

Factor	Slow speed engine	High speed engine
Break mean effective pressure - therefore wear and tear and consumption of spares	Low	High
Weight to power ratio- therefore sturdiness and life	More	Less
Space	High	Less
Type of use	Continuous use	Intermittent use
Period between overhauls*	8000 hours	3200
Direct operating cost (includes lubricating oils, filters etc.	Less	High

* Typical recommendations from manufacturers

Keeping the above factors and available capacities of DG set in mind, the cost of economics for both the engines should be worked out before arriving at a decision.

Capacity Combinations

From the point of view of space, operation, maintenance and initial capital investment, it is certainly economical to go in for one large DG set than two or more DG sets in parallel.

Two or more DG sets running in parallel can be a advantage as only the short-fall in power –depending upon the extent of power cut prevailing - needs to filled up. Also, flexibility of operation is increased since one DG set can be stopped, while the other DG set is generating at least 50% of the power requirement. Another advantage is that one DG set can become 100% standby during lean and low power-cut periods.

Air Cooling Vs. Water Cooling

The general feeling has been that a water cooled DG set is better than an air cooled set, as most users are worried about the overheating of engines during summer months. This is to some extent is true and precautions have to be taken to ensure that the cooling water temperature does not exceed the prescribed limits. However, from performance and maintenance point of view, water and air cooled sets are equally good except that proper care should be taken to ensure cross ventilation so that as much cool air as possible is circulated through the radiator to keep its cooling water temperature within limits.

While, it may be possible to have air cooled engines in the lower capacities, it will be necessary to go in for water cooled engines in larger capacities to ensure that the engine does not get over-heated during summer months.

Safety Features

It is advisable to have short circuit, over load and earth fault protection on all the DG sets. However, in case of smaller capacity DG sets, this may become uneconomical. Hence, it is strongly recommended to install a circuit protection. Other safety equipment like high temperature, low lube oil pressure cut-outs should be provided, so that in the event of any of these abnormalities, the engine would stop and prevent damage. It is also essential to provide reverse power relay when DG sets are to run in parallel to avoid back feeding from one alternator to another.

Parallel Operation with Grid

Running the DG set in parallel with the mains from the supply undertakings can be done in consultation with concerned electricity authorities. However, some supply undertakings ask the consumer to give an undertaking that the DG set will not be run in parallel with their supply. The reasons stated are that the grid is an infinite bus and paralleling a small capacity DG set would involve operational risks despite normal protections like reverse power relay, voltage and frequency relays.

Maximum Single Load on DG Set

The starting current of squirrel cage induction motors is as much as six times the rated current for a few seconds with direct-on-line starters. In practice, it has been found that the starting current value should not exceed 200 % of the full load capacity of the alternator. The voltage and frequency throughout the motor starting interval recovers and reaches rated values usually much before the motor has picked up full speed.

In general, the HP of the largest motor that can be started with direct on line starting is about 50 % of the kVA rating of the generating set. On the other hand, the capacity of the induction motor can be increased, if the type of starting is changed over to star delta or to auto transformer starter, and with this starting the HP of the largest motor can be upto 75 % of the kVA of Genset.

Unbalanced Load Effects

It is always recommended to have the load as much balanced as possible, since unbalanced loads can cause heating of the alternator, which may result in unbalanced output voltages. The maximum unbalanced load between phases should not exceed 10 % of the capacity of the generating sets.

Neutral Earthing

The electricity rules clearly specify that two independent earths to the body and neutral should be provided to give adequate protection to the equipment in case of an earth fault, and also to drain away any leakage of potential from the equipment to the earth for safe working.

Site Condition Effects on Performance Derating

Site condition with respect to altitude, intake temperature, cooling water temperature and derate diesel engine output as shown in following Tables: 9.3 and 9.4.

Table 9.3 Altitude and Intake Temperature Corrections

Correction Factors For Engine Output				
Altitude Correction			Temperature Correction	
Altitude Meters over MSL	Non Super Charged	Super Charged	Intake °C	Correction Factor
610	0.980	0.980	32	1.000
915	0.935	0.950	35	0.986
1220	0.895	0.915	38	0.974
1525	0.855	0.882	41	0.962
1830	0.820	0.850	43	0.950
2130	0.780	0.820	46	0.937
2450	0.745	0.790	49	0.925
2750	0.712	0.765	52	0.913
3050	0.680	0.740	54	0.900
3660	0.612	0.685		
4270	0.550	0.630		
4880	0.494	0.580		

Table 9.4 Derating due to Air Inter Cooler Water Inlet		
Water Temperature °C	**Flow %**	**Derating %**
25	100	0
30	125	3
35	166	5
40	166	8

9.3 Operational Factors

Load Pattern & DG Set Capacity

The average load can be easily assessed by logging the current drawn at the main switchboard on an average day. The 'over load' has a different meaning when referred to the D.G. set. Overloads, which appear insignificant and harmless on electricity board supply, may become detrimental to a D.G.set, and hence overload on D.G.set should be carefully analysed. Diesel engines are designed for 10% overload for 1 hour in every 12 hours of operation. The A.C. generators are designed to meet 50% overload for 15 seconds as specified by standards. The D.G.set/s selection should be such that the overloads are within the above specified limits. It would be ideal to connect steady loads on DG set to ensure good performance. Alongside alternator loading, the engine loading in terms of kW or BHP, needs to be maintained above 50%. Ideally, the engine and alternator loading conditions are both to be achieved towards high efficiency.

Engine manufacturers offer curves indicating % Engine Loading vs fuel Consumption in grams/BHP. Optimal engine loading corresponding to best operating point is desirable for energy efficiency.

Alternators are sized for kVA rating with highest efficiency attainable at a loading of around 70% and more. Manufacturer's curves can be referred to for best efficiency point and corresponding kW and kVA loading values.

Sequencing of Loads

The captive diesel generating set has certain limits in handling the transient loads. This applies to both kW (as reflected on the engine) and kVA (as reflected on the generator). In this context, the base load that exists before the application of transient load brings down the transient load handling capability, and in case of A.C. generators, it increases the transient voltage dip. Hence, great care is required in sequencing the load on D.G.set/s. It is advisable to start the load with highest transient kVA first followed by other loads in the descending order of the starting kVA. This will lead to optimum sizing and better utilisation of transient load handling capacity of D.G.set.

Load Pattern

In many cases, the load will not be constant throughout the day. If there is substantial variation in load, then consideration should be given for parallel operation of D.G.sets. In such a situation, additional D.G. set(s) are to be switched on when load increases. The typical case may be an establishment demanding substantially different powers in first, second and third shifts. By parallel operation, D.G. sets can be run at optimum operating points or near about, for optimum fuel consumption and

additionally, flexibility is built into the system. This scheme can be also be applied where loads can be segregated as critical and non-critical loads to provide standby power to critical load in the captive power system.

Load Characteristics

Some of the load characteristics influence efficient use of D.G.set. These characteristics are entirely load dependent and cannot be controlled by the D.G.set. The extent of detrimental influence of these characteristics can be reduced in several cases.

- **Power Factor:**

The load power factor is entirely dependent on the load. The A.C. generator is designed for the power factor of 0.8 lag as specified by standards. Lower power factor demands higher excitation currents and results in increased losses. Over sizing A.C. generators for operation at lower power factors results in lower operating efficiency and higher costs. The economical alternative is to provide power factor improvement capacitors.

- **Unbalanced Load:**

Unbalanced loads on A.C. generator leads to unbalanced set of voltages and additional heating in A.C. generator. When other connected loads like motor loads are fed with unbalanced set of voltages additional losses occur in the motors as well. Hence, the load on the A.C. generators should be balanced as far as possible. Where single phase loads are predominant, consideration should be given for procuring single phase A.C. generator.

- **Transient Loading:**

On many occasions to contain transient voltage dip arising due to transient load application, a specially designed generator may have to be selected. Many times an unstandard combination of engine and A.C. generator may have to be procured. Such a combination ensures that the prime mover is not unnecessarily over sized which adds to capital cost and running cost.

- **Special Loads:**

Special loads like rectifier / thyristor loads, welding loads, furnace loads need an application check. The manufacturer of diesel engine and AC generator should be consulted for proper recommendation so that desired utilisation of DG set is achieved without any problem. In certain cases of loads, which are sensitive to voltage, frequency regulation, voltage wave form, consideration should be given to segregate the loads, and feed it by a dedicated power supply which usually assumes the form of DG motor driven generator set. Such an alternative ensures that special design of AC generator is restricted to that portion of the load which requires high purity rather than increasing the price of the D.G.set by specially designed AC generator for complete load.

Waste Heat Recovery in DG Sets

For combined heat and power applications, waste heat from reciprocating engines can be tapped mainly from exhaust gases and cooling water that circulates around cylinders in the engine jackets, with

additional potential from oil and turbo coolers. While engine exhaust and cooling water each provide about half of the useful thermal energy, the exhaust is at much higher temperature (around 450 °C versus 100 °C) and hence is more versatile. A typical energy balance in a reciprocating engine generator using Diesel and Natural gas is given in Table 9.5 below.

Table 9.5 Energy Balance for Reciprocating Engine

	Conventional cooling system		Cooling system with engine jacket and exhaust heat recovery	
500-kW natural gas engine generator*				
Electric power	**30%**		**30%**	
Jacket-water heat	**38%**		**38%**	**54% recoverable**
Exhaust heat	**24%**	**70% wasted**	**Exh recoverable 16%**	
			Exh lost 8%	**16% wasted**
Radiated heat lost to atmosphere	**8%**		**8%**	
	100%		**100%**	
500-kW diesel engine generator*				
Electric power	**35%**		**35%**	
Jacket water	**32%**		**32%**	**48% recoverable**
Exhaust heat	**24%**	**65% wasted**	**Exh recoverable 16%**	
			Exh lost 8%	**17% wasted**
Radiated heat lost to atmosphere	**9%**		**9%**	
	100%		**100%**	

The table reveals that for natural gas generator more thermal energy (54%) can be recovered from the reciprocating engine compared to an electrical energy conversion of 30%.

It would be realistic to assess the Waste Heat Recovery (WHR) potential in relation to quantity, temperature margin, in kcal/hr as:

$$\text{Potential WHR} = (\text{kWh Output/hour}) \times (8 \text{ kg Gases / kWh Output}) \times 0.25 \text{ kcal/kg}^{o}\text{C} \times (t_g - 180^{o}\text{C})$$

Where, t_g is the gas temperature after Turbocharger, (the criteria being that limiting exit gas temperature cannot be less than 180°C, to avoid acid dew point corrosion), 0.25 being the specific heat of flue gases and kWh output being the actual average unit generation from the set per hour. For a 1100 kVA set, at 800 kW loading, and with 480°C exhaust gas temperature, the waste heat potential works out to:

$$800 \text{ kWh} \times 8 \text{ kg gas generation / kWh output} \times 0.25 \text{ kcal/kg }^{o}\text{C} \times (480 - 180), \text{ i.e., } 4,80,000 \text{ kcal/hr}$$

While the above method yields only the potential for heat recovery, the actual realisable potential depends upon various factors and if applied judiciously, a well configured waste heat recovery system can tremendously boost the economics of captive DG power generation.

The factors affecting Waste Heat Recovery from flue Gases are:

a) DG Set loading, temperature of exhaust gases

b) Hours of operation and

c) Back pressure on the DG set

Consistent DG set loading (to over 60% of rating) would ensure a reasonable exit flue gas quantity and temperature. Fluctuations and gross under loading of DG set results in erratic flue gas quantity and temperature profile at entry to heat recovery unit, thereby leading to possible cold end corrosion and other problems. Typical flue gas temperature and flow pattern in a 5 MW DG set at various loads are given in Table 9.6.

Table 9.6 Typical Flue Gas Temperature and Flow Pattern in a 5-MW DG Set at various Loads

100% Load	11.84 kg/sec	370°C
90% Load	10.80 kg/sec	350°C
70% Load	9.08 kg/sec	330°C
60% Load	7.50 kg/sec	325°C
If the normal load is 60%, the flue gas parameters for waste heat recovery unit would be 325°C inlet temperature, 180°C outlet temperature and 27180 kg/hour gas flow. At 90% loading, however, values would be 350°C and 32,400 kgs/Hour, respectively.		

* Number of hours of operation of the DG Set has an influence on the thermal performance of waste heat Recovery unit. With continuous DG Set operations, cost benefits are favourable.
* Back pressure in the gas path caused by additional pressure drop in waste heat recovery unit is another key factor. Generally, the maximum back pressure allowed is around 250-300 mmWC and the heat recovery unit should have a pressure drop lower than that. Choice of convective waste heat recovery systems with adequate heat transfer area are known to provide reliable service.

The configuration of heat recovery system and the choice of steam parameters can be judiciously selected with reference to the specific industry (site) requirements. Much good work has taken place in Indian Industry regarding waste heat recovery and one interesting configuration, deployed is installation of waste heat boiler in flue gas path along with a vapour absorption chiller, to produce 8°C chilled water working on steam from waste heat.

Trigeneration Technology

In order to further optimize fuel utilization Trigeneration systems are developed which involves the simultaneous production of electricity, heat and cooling. The prime mover used for power generation includes diesel engines/gas engines. The waste heat recovery system in captive power generation units consists of waste heat recovery boiler for generating steam and use of jacket cooling water for operating Vapor Absorption Machines (VAM) to meet Air conditioning requirements.

9.4 Energy Performance Assessment of DG Sets

Routine energy efficiency assessment of DG sets on shop floor involves following typical steps:

1) Ensure reliability of all instruments used for trial.
2) Collect technical literature, characteristics, and specifications of the plant.

3) Conduct a 2 hour trial on the DG set, ensuring a steady load, wherein the following measurements are logged at 15 minutes intervals.
   a) Fuel consumption (by dip level or by flow meter)
   b) Amps, volts, PF, kW, kWh
   c) Intake air temperature, Relative Humidity (RH)
   d) Intake cooling water temperature
   e) Cylinder-wise exhaust temperature (as an indication of engine loading)
   f) Turbocharger RPM (as an indication of loading on engine)
   g) Charge air pressure (as an indication of engine loading)
   h) Cooling water temperature before and after charge air cooler (as an indication of cooler performance)
   i) Stack gas temperature before and after turbocharger (as an indication of turbocharger performance)

4) The fuel oil/diesel analysis is referred to from an oil company data.

5) Analysis: The trial data is to be analysed with respect to:
   a) Average alternator loading.
   b) Average engine loading.
   c) Percentage loading on alternator.
   d) Percentage loading on engine.
   e) Specific power generation kWh/liter.
   f) Comments on Turbocharger performance based on RPM and gas temperature difference.
   g) Comments on charge air cooler performance.
   h) Comments on load distribution among various cylinders (based on exhaust temperature, the temperature to be ± 5% of mean and high/low values indicate disturbed condition).
   i) Comments on housekeeping issues like drip leakages, insulation, vibrations, etc.

A format as shown in the Table 9.7 is useful for monitoring the performance

Table 9.7 Typical Format for DG Set Monitoring

DG Set No.	Electricity Generating Capacity (Site), kW	Derated Electricity Generating Capacity, kW	Type of Fuel used	Average Load as % of Derated Capacity	Specific Fuel Cons. Lit/ kWh	Specific Lube Oil Cons. Lit/kWh
1.	480	300	LDO	89	0.335	0.007
2.	480	300	LDO	110	0.334	0.024
3.	292	230	LDO	84	0.356	0.006
4.	200	160	HSD	89	0.325	0.003
5.	200	160	HSD	106	0.338	0.003
6.	200	160	HSD			
7.	292	230	LDO	79	0.339	0.006
8.	292	230	LDO	81	0.362	0.005
9.	292	230	LDO	94	0.342	0.003
10.	292	230	LDO	88	0.335	0.006
11.	292	230	LDO	76	0.335	0.005
12.	292	230	LDO	69	0.353	0.006

Table 9.7 Typical Format for DG Set Monitoring						
DG Set No.	**Electricity Generating Capacity (Site), kW**	**Derated Electricity Generating Capacity, kW**	**Type of Fuel used**	**Average Load as % of Derated Capacity**	**Specific Fuel Cons. Lit/ kWh**	**Specific Lube Oil Cons. Lit/kWh**
13	400	320	HSD	75	0.334	0.004
14.	400	320	HSD	65	0.349	0.004
15.	880	750	LDO	85	0.318	0.007
16.	400	320	HSD	70	0.335	0.004
17.	400	320	HSD	80	0.337	0.004
18.	880	750	LDO	78	0.345	0.007
19.	800	640	HSD	74	0.324	0.002
20.	800	640	HSD	91	0.290	0.002
21.	880	750	LDO	96	0.307	0.002
22.	920	800	LDO	77	0.297	0.002

9.5 Energy Saving Measures for DG Sets

a) Ensure steady load conditions on the DG set, and provide cold, dust free air at intake (use of air washers for large sets, in case of dry, hot weather, can be considered).

b) Improve air filtration.

c) Ensure fuel oil storage, handling and preparation as per manufacturer's guidelines/oil company data.

d) Consider fuel oil additives in case they benefit fuel oil properties for DG set usage.

e) Calibrate fuel injection pumps frequently.

f) Ensure compliance with maintenance checklist.

g) Ensure steady load conditions, avoiding fluctuations, imbalance in phases, harmonic loads.

h) In case of a base load operation, consider waste heat recovery system adoption for steam generation or refrigeration chiller unit incorporation. Even the Jacket Cooling Water is amenable for heat recovery, vapour absorption system adoption.

i) In terms of fuel cost economy, consider partial use of biomass gas for generation. Ensure tar removal from the gas for improving availability of the engine in the long run.

j) Consider parallel operation among the DG sets for improved loading and fuel economy thereof.

k) Carryout regular field trials to monitor DG set performance, and maintenance planning as per requirements.

Solved Example:

a) A 180 kVA, 0.80 PF rated DG set has diesel engine rating of 210 BHP. What is the maximum power factor which can be maintained at full load on the alternator without overloading the DG set? (Assume alternator losses and exciter power requirement as 5.66 kW and there is no derating of DG set)

Ans:

Engine rated Power	=	210 x 0.746	=	156.66 kW
Rated power available for alternator	=	156.66 – 5.66	=	151 kW
Maximum power factor possible	=	151 /180	=	0.84

b) A DG set is operating at 600 kW load with 450^{O}C exhaust gas temperature. The DG set generates 8 kg of exhaust gas/ kWh generated. The specific heat of gas at 450^{o}C is 0.25 kcal/ kg^{O}C. A heat recovery boiler is installed after which the exhaust temperature drops to 230^{O}C. How much steam will be generated at 3 kg/ cm^{2} with enthalpy of 650.57 kcal/ kg. Assume boiler feed water temperature as 80^{o}C.

Ans:

Waste Heat Recovery = 600 kWh x 8 kg gas generated/ kWh output x 0.25 kcal/ kg ^{o}C x (450^{o}C-230 ^{o}C) =2,64,000 kcal/hr

Steam generation = 2,64,000 kCal/hr / (650.57 – 80) = 462.7 kg/ hr.

QUESTIONS	
Objective Type Questions	
1.	The compression ratio in diesel engines is in the range of: a) 10:1 to 15:1 b) 14:1 to 25:1 c) 5:1 to 10:1 d) 1:2 to 3:1
2.	Present specific fuel consumption value of DG sets in industries is about _____. a) 220 g/kWh b) 100 g/kWh c) 160 g/kWh_ d) 50 g/kWh
3.	The rating required for a DG set with 500 kW connected load and with diversity factor of 1.5, 80% loading and 0.8 power factor is _____ a) 520 kVA b) 600 kVA c) 625 kVA d) 500 kVA
4.	The waste heat potential for a 1100 kVA set at 800 kW loading and with 480 °C exhaust gas temperature is _____ a) 4.8 lakh kcal/hr b) 3.5 lakh kcal/hr c) 3 lakh kcal/hr d) 2 lakh kcal/hr
5.	For a DG set, the copper losses in the alternator are proportional to the: a) current delivered by the alternator b) square of the current delivered by the alternator c) square root of the current delivered by the alternator d) none of the above
6.	The capacity of largest motor that can be started in the given DG set is a) 25% b) 50% kVA rating of DG set c) 75% d) 100%
7.	The jacket cooling water in a diesel engine flows at 12.9 m^3/hr with a range of 10°C and accounts for 30% of the engine input energy. The power loss in cooling water is a) 430 kW b) 500 kW c) 387 kW d) 150 kW
8.	The operating economics of trigeneration primarily depends on the a) capacity of generator b) capacity of waste heat boiler c) capacity of VAM d) cost of fuel
9.	The efficiency of a Diesel Generator Power plant ranges between: a) 20 – 25% b) 0 – 20% c) 40 – 45% d) 60 – 70%
10.	A spark ignition engine is used for firing which of the following fuels a) high speed diesel b) light diesel oil c) natural gas d) furnace oil
Short Type Questions	
S-1	A 5 MW DG set with an average load of 3 MW running in parallel with the grid was found to be exporting 100 kVAr. Without calculating, explain what could be the possible reasons for the export of reactive power to the grid. List advantages/disadvantages of the above situation.

S-2	A DG set is operating at 750 kW loading with 440^{o}C exhaust gas temperature. The DG set generates 8 kg gas/ kWh generated, and specific heat of gas at 0.25 kcal/ kg ^{o}C. A heat recovery boiler is installed after which the exhaust gas temperature reduces to 190^{o}C. How much steam will be generated at 3 kg/ cm^{2} with enthalpy of 650 kcal/ kg. Assume boiler feed water temperature as 70^{o}C.
S-3	The connected load of a plant is 1200 kW and Diversity factor is 1.8. What is the desirable set rating with respect to 0.8 PF and the set load factor of 75%?
S-4	What are the sources of waste heat recovery in a DG set?. Waste heat recovery from which source is more economically attractive. Why?
S-5	Explain the principle of a four stroke diesel engine?
Long Type Questions	
L-1	A cement industry has installed 6 MW DG set and is operated by furnace oil. The cost of furnace oil is Rs.28 / liter. The average loading on the DG set is 5 MW and the hourly furnace oil consumption is 1230 liters. Estimate the cost of power generation in Rs./kWh. The plant management is planning to convert the existing DG set to gas operated DG set. The estimated cost of gas is Rs.20/Sm^{3}. If the power generation is 3.7 kWh / Sm^{3}. Calculate the generation cost per kWh with gas as fuel.
L-2	The process plant has installed 3.5 MW DG sets. Furnace oil is used as a fuel in DG sets. The load factor of the DG sets 80% and DG set is operated for 7000 hours in a year. Calculate the energy generated and the waste heat recovery potential from exhaust gas.

REFERENCES

1. Proceedings of National Workshop on Efficient Captive Power Generation with Industrial DG Sets
2. NPC Case Studies

Chapter 10

Energy Conservation in Buildings and ECBC

Learning Objectives

In this chapter you will learn about

- Building definition
- Energy conservation building codes (ECBC)
- ECBC guidelines on building envelope
- ECBC guidelines on heating ventilation and air conditioning
- ECBC guidelines on service hot water
- Water pumping system
- ECBC guidelines on lighting
- Uninterruptible power supply
- Escalators and elevators
- Building management system
- Star rating of buildings

10. ENERGY CONSERVATION IN BUILDINGS AND ECBC

About ECBC, Building envelope, Fenestrations, Insulation, HVAC, Lighting, Water pumping, Inverters and energy storage / captive generation, Elevators and escalators, Star labeling for existing buildings.

10.1 Introduction

There are several different uses of energy in buildings. The major uses are for lighting, heating, cooling, power delivery to equipment and appliances, and domestic water. The amount that each contributes to the total energy use varies according to the climate, type of building, number of working hours and time of year. Energy use for air-conditioning has the largest share at a national level. In areas where severe winters occur, heating load will be greater than cooling load in terms of the total energy use. In some types of buildings in certain climatic zones, the lighting load might be greater than either the heating or cooling loads.

Industrial and commercial buildings are dissimilar in terms of energy use, as industries primarily use large quantities of energy for specialized processes whereas buildings use the major amount of energy for human comfort. It is difficult to generalize energy use by type of building because there are many variables that determine the energy use in a particular building.

10.2 Building Definition as in the Energy Conservation (amendment) Act 2010

"building" means any structure or erection or part of structure or erection after the rules relating to energy conservation building codes have been notified under clause (*p*) of section 14 and clause (*a*) of section 15 and includes any existing structure or erection or part of structure or erection, which is having a connected load of 100 Kilowatt (kW) or contract demand of 120 Kilovolt Ampere (kVA) and above and is used or intended to be used for commercial purposes.

10.3 Energy Conservation Building Code (ECBC)

Energy Conservation Building Code defines the minimum energy efficiency standards for design and construction of commercial buildings, to encourage energy efficient design or major retrofit of buildings without any compromise with the building function, comfort, health, or the productivity of the occupants.

In order to implement ECBC across the country, India has been divided into five climatic zones as per the weather conditions (Figure 10.1). The Five Climatic zones are:

- Composite
- Hot Dry

- Warm Humid
- Moderate
- Cold

The ECBC Building Code considers the following aspects of the Buildings:

a) Building envelopes, except for unconditioned storage spaces or warehouses,

b) Mechanical systems and equipment, including heating, ventilation, and air conditioning,

c) Service hot water heating,

d) Interior and exterior lighting, and

e) Electrical power and motors.

The provisions of this ECBC code do not apply to:

a) Buildings that do not use either electricity or fossil fuel,

b) Equipment and portions of building systems that use energy primarily for manufacturing processes, and

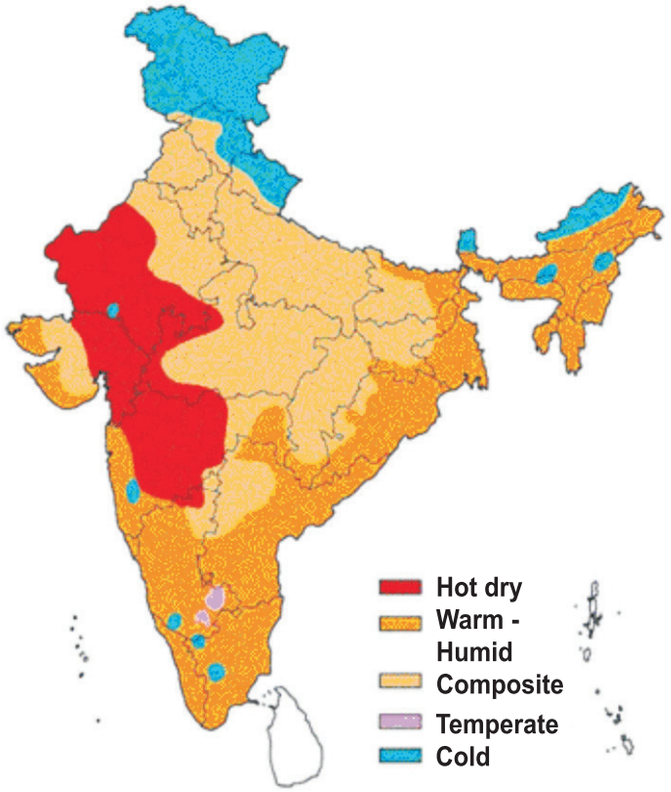


Figure 10.1 Five Climatic Zones as per the Weather Conditions

10.4 Compliance Approaches

The Code requires that the building shall comply first with all the mandatory provisions discussed in Chapter 4 to 8 (of the Code). But every building project is different: each building has its own site that presents unique opportunities and challenges, each building owner or user has different requirements, and climate and microclimate conditions can vary significantly among projects. Architects and engineers need flexibility in order to design buildings that address these diverse requirements. The Code provides this flexibility in a number of ways. Building components and systems have multiple options to comply with the Code requirements. To use the building envelope section as an example, designers can choose the Prescriptive Method that requires roof insulation be installed with a minimum R-value. Alternatively, the other options allow the designer to show compliance with the thermal performance (U-factor) of roof construction assembly. In addition building envelope tradeoff option discussed in Chapter 4 permits trade-offs among building envelope components (roof, walls, and fenestration) for Code compliance. If more flexibility is needed, the Whole Building Performance (WBP) Method approach may be used, but WBP requires simulation expertise.

a. Prescriptive Method

The Code specifies a set of prescriptive requirements for building systems and components. Compliance with the Code can be achieved by meeting or exceeding the specific levels described for each individual element of the building systems, covered in Chapter 4 through Chapter 8 of the Code. For building envelope, the Code provides a Trade-Off option that allows trading off the efficiency of one envelope element with another to achieve the overall efficiency level required by the Code. The envelope trade-off option is discussed in Chapter 12: Appendix D of ECBC.

b. Whole Building Performance Method

Use of energy simulation software is necessary to show ECBC compliance via the Whole Building Performance Method. Energy simulation is a computer-based analytical process that helps building owners and designers to evaluate the energy performance of a building and make it more energy-efficient by making necessary modifications in the design before the building is constructed.

These computer-based energy simulation programs model the thermal, visual, ventilation, and other energy consuming processes taking place within the building to predict its energy performance. The simulation program takes into account the building geometry and orientation, building materials, building façade design and characteristics, climate, indoor environmental conditions, occupant activities and schedules, HVAC and lighting system and other parameters to analyze and predict the energy performance of the building. Computer simulation of energy use can be accomplished with a variety of computer software tools and in many cases may be the best method for guiding a building project to be energy-efficient. However, this approach does require considerable knowledge of building simulation tools and very close communication between members of the design team.

Appendix B of the Code describes the Whole Building Performance Method for complying with the Code. This method involves developing a computer model of the Proposed Design and comparing its energy consumption to the Standard Design for that building. Energy consumption in the Standard Design represents the upper limit of energy use allowed for that particular building under a scenario where all the prescriptive requirements of the Code are adopted. Code compliance will be achieved if the energy use in Proposed Design is no greater than the energy used in the Standard Design.

Three basic steps are involved:

1. Design the building with energy efficiency measures; the prescriptive approach requirements provide a good starting point for the development of the design.
2. Demonstrate that the building complies with the mandatory measures.
3. Using approved simulation software, model the energy consumption of the building using the proposed features to create the Proposed Design. The model will also automatically calculate the energy use for the Proposed Design.

If the energy use in Proposed Design is lesser than the energy use in the Standard Design, the building complies with the Code.

10.5 ECBC Guidelines on Building Envelope

The building envelope refers to the exterior façade, and is comprised of walls, windows, roof, skylights, doors, and other openings. The envelope protects the building's interior and occupants from the weather conditions and other external elements. The design features of the envelope strongly affect the visual and thermal comfort of the occupants, as well as energy consumption in the building.

Building Envelope

The Exterior and semi-exterior portions in the context of defining as building envelope include:

- Elements that separate the conditioned spaces from the weather conditions, or
- Elements of a building that separate the conditioned spaces of the building from the unconditioned spaces, i.e. office space from unconditioned storage.

Building Envelope Sealing

The following areas of the enclosed building envelope shall be sealed, caulked, gasketed, or weather-stripped to minimize air leakage:

a) Joints around fenestration and door frames,
b) Openings between walls and foundations and between walls and roof and wall panels,
c) Openings at penetrations of utility services through, roofs, walls, and floors
d) Site-built fenestration and doors,
e) Building assemblies used as ducts or plenums, and
f) All other openings in the building envelope.

Envelope design Basics

From an energy efficiency point of view, the envelope design must take into consideration both the external and internal heat loads, as well as day lighting benefits. External loads include mainly solar heat gains through windows, heat losses across the envelope surfaces, and infiltration in the building. Internal loads include heat gain from electric lighting systems, equipment, and people working in the building.

One of the goals of the envelope design should be to introduce day lighting into the interior space of the building through windows and skylights, thereby reducing the need for electric lighting. Thus, giving proper orientation to the building and due consideration to the size and placement of windows at the design stage can provide the advantage of day lighting.

Secondly, to maintain thermal comfort and minimize internal cooling / heating loads, the building envelope needs to regulate and optimize heat transfer through roof, walls, windows, skylights, doors and other openings. Effective insulation of roof and walls, appropriate selection of glazing and framing for windows, and suitable shading strategy are important in designing energy efficient buildings.

Passive Solar Design Strategy

Architects should pay attention to the following basic design elements in an effort to reduce the energy consumption in small commercial buildings that can be operated without Central HVAC System.

Siting and Orientation

In a predominantly hot climate, cooling load affects the total energy consumption in commercial buildings in a significant manner. Thus, controlling heat transfer through the roof, walls, and windows becomes of utmost importance and needs to be considered from the initial stages of design. Therefore, site planners and designers should properly orient buildings to minimize solar gains in the summer.

Plot lines and roads should be situated to minimize building exposure to the east and west. These orientations provide the highest solar heat gains. Subdivisions should be planned so that the longer sides of the buildings face north and south. With proper planning, there may be no added costs for good orientation.

Shade

Use different shading strategies to minimize solar heat gain and reduce glare inside buildings. Provide vertical louvers on east and west side and horizontal shading devices on south side.

Cross-Ventilation

Building envelope should allow the movement of breeze throughout the building.

Cool Roof

This refers to the property of roof that describes its ability to reflect and reject heat. Cool roof (Figure 10.2) surfaces have both high solar reflectance and a high emittance (reject heat back to the environment).

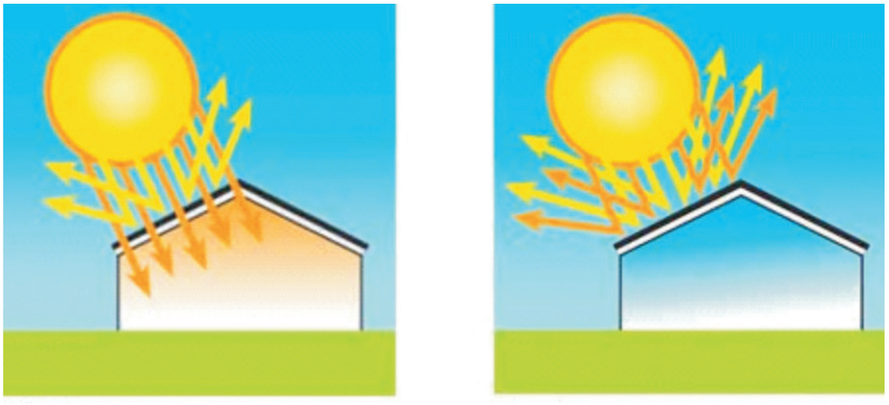


Figure 10.2 (a) Conventional Roof (b) Cool Roof

Solar Reflectance and Absorptance

The solar reflectance is the fraction of solar radiation reflected by roof. The complement of reflectance is absorptance (Figure 10.3); whatever radiant energy incident on a surface that is not reflected is absorbed in the roof. The reflectance and absorptance of building materials are usually measured across the solar spectrum, since these are exposed to that range of wavelength. Reflectance is measured on a scale of 0 to 1, with 0 being a perfect absorber and 1 being a perfect reflector. Absorptance is also rated from 0 to 1, and can be calculated from the relation: Reflectance + Absorptance = 1.

Thermal Emittance

Thermal emittance is the relative ability of a material to radiate the absorbed heat (Figure 10.3). Emissivity (or thermal emittance) of a material is the ratio of energy radiated by a particular material to energy radiated by a black body at the same temperature. It is a measure of a material's ability to radiate the absorbed energy.

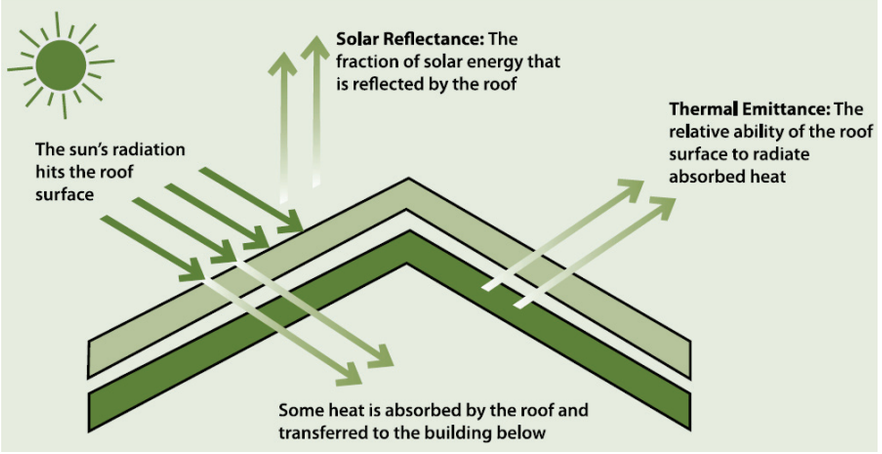


Figure 10.3 Solar Reflectance, Absorptance & Thermal Emittance

A true black body would have an e =1 while any real object would have e <1. Emissivity is a dimensionless quantity (does not have units). In general, the duller and blacker a material is, the closer its emissivity is to 1. The more reflective a material is, the lower its emissivity. The emissivity of building material, unlike reflectance, is usually measured in the far infrared part of the spectrum.

Solar Heat Gain Coefficient (SHGC)

Regardless of outside temperature, heat can be gained through windows by direct or indirect solar radiation. The ability to control this heat gain through windows is characterized in terms of the SHGC of the window. SHGC is the ratio of solar heat gain that passes through fenestration to the total incident solar radiation on the fenestration (Figure 10.4). Solar heat gains includes directly transmitted solar heat and absorbed solar heat, which is re-radiated, conducted, on convected into the interior space.

Visible Light Transmittance (VLT)

It is the ratio of light passing through the glazing to light passing through perfectly transmissive glazing (Figure 10.4). VLT is concerned only with the visible portion of the solar spectrum, as opposed to SHGC, which is the ratio of all solar radiation. VLT is an important parameter for day lighting of buildings.

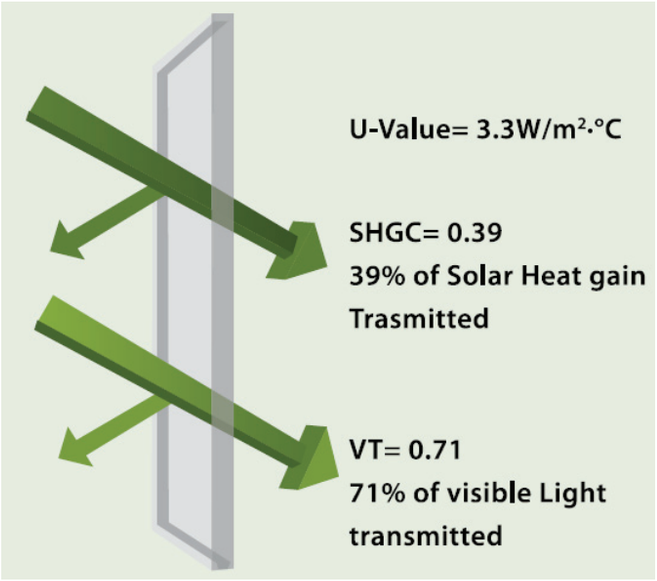


Figure 10.4 Solar Heat Gain Coefficient & Visible Light Transmittance

Fenestration

Fenestration systems include windows, skylights, ventilators, and doors that are more than one-half glazed. All openings (including the frames) in the building envelope that let in light.

Skylight

A fenestration surface having a slope of less than 60 degrees from the horizontal plane (Figure 10.5).

Figure 10.5 Skylight Roof Window

Vertical Fenestration

All fenestration other than skylights.

Fenestration Area

Total area of the fenestration measured using the rough opening (including glazing, sash and frame). For glass doors where glazed vision area is less than 50% of the door area, the fenestration area is the glazed vision area; otherwise, it is the door area.

Window-Wall Ratio (WWR)

Ratio of vertical fenestration area to gross exterior wall area. Gross exterior wall area is measured horizontally from the exterior surface; it is measured vertically from the top of the floor to the bottom of the roof.

Effective Aperture of Glazing

In simple terms, as the area of an aperture /opening in the building envelope increases, the amount of daylight received in the building space also increases. However the glazing material within that aperture can effectively reduce the amount of visible light that enters the space. Therefore, aperture size alone is not an effective determinant to measure illumination levels. If the glazing in an opening is perfectly transparent material the effective aperture size would be equal to the area of opening (because the visible transmittance of the glazing would be one). If however the glazing has a VLT of 0.5, the opening will transmit only half of the light striking it, and the effective aperture will be half of the actual size of the opening.

The Effective Aperture (EA) or light admitting potential of a glazing system is determined by multiplying the Visible Light Transmittance of the glazing by the window-to-wall ratio (WWR) of the building. The window to wall ratio is the net window area to the exterior wall area.

This value is given as equal to Visible Light Transmittance X Window-Wall Ratio

$$EA = VLT \times WWR$$

Example:

Case 1:

WWR = 0.4

VLT = 0.26

EA = 0.4x0.26 = 0.104

EA > 0.1 Glazing complies with ECBC

Case 2:

WWR = 0.6

VLT = 0.15

EA = 0.6x0.15 = 0.09

EA < 0.1 Glazing does not complies with ECBC

The value of VLT is taken from the ECBC for different Window to Wall Ratio (WWR).

Opaque Wall:

All areas in the building envelope, except fenestration and building service openings such as vents and grills.

Sound Transmission

An important requirement in some type of projects like theaters, hotels, where Energy efficient glazing delivers improved acoustic performance as a side benefit.

Spectral Selectivity

This refers to the ability of a glazing material to respond differently to different wavelengths of solar energy. In other words, to admit visible light while rejecting unwanted invisible infrared heat. Newer glazing products have achieved this characteristic, permitting much clearer glass than previously available for solar control glazing. A glazing with a relatively high Visible Light Transmittance (VLT) and a low Solar Heat Gain Coefficient (SHGC) indicates that a glazing use special absorbing tints or coatings, and are typically either clear or have a blue or blue/green appearance.

Weather stripping

Materials, such as a strip of fabric, plastic, rubber or metal, or a device used to seal the openings, gaps or cracks of venting window and door units to prevent water and air infiltration.

U-Factor (W/m^2 K)

When there is a temperature difference between inside and outside, heat is lost or gained through the window frame and glazing by the combined effects of conduction, convection, and long wave radiation. U-factor is the rate of heat flow through one square meter of wall/fenestration assembly when there is 1°C temperature difference. The lower the U-factor, the lesser heat transfer takes place. Center-of-glass U-factors are generally lower than whole-window U-factors, which account for the effect of the frame and mullions. This property is important for reducing heating load in cold climates and for reducing cooling load in hot climates.

Roof and opaque wall's U-factor also refers to the amount of heat transferred (lost/gain), due to a temperature differential of 1°C between inside and outside, per square meter.

The overall heat transfer coefficient and the insulation resistance for walls and roofs are indicated in Table 10.1 & 10.2. The vertical fenestration U factor and SGHC requirements as per ECBC guidelines are given in Table 10.3.

Table 10.1: Opaque Wall Assembly U-factor and Insulation R-value Requirements

Climate Zone	Hospitals, Hotels, Call Centers (24-Hour)		Other Building Types (Daytime)	
	Maximum U-factor of the overall assembly (W/m^2-°C)	Minimum R-value of insulation alone (m^2-°C/W)	Maximum U-factor of the overall assembly (W/m^2-°C)	Minimum R-value of insulation alone (m^2-°C/W)
Composite	U-0.440	R-2.10	U-0.440	R-2.10
Hot and Dry	U-0.440	R-2.10	U-0.440	R-2.10
Warm and Humid	U-0.440	R-2.10	U-0.440	R-2.10
Moderate	U-0.431	R-1.80	U-0.397	R-2.00
Cold	U-0.369	R-2.20	U-0.352	R-2.35

Table 10.2: Roof assembly U-factor and Insulation R-value Requirements

Climate Zone	Hospitals, Hotels, Call Centers (24-Hour)		Other Building Types (Daytime)	
	Maximum U-factor of the overall assembly ($W/m^2 °C$)	Minimum R-value of insulation alone ($m^2 °C/W$)	Maximum U-factor of the overall assembly ($W/m^2 °C$)	Minimum R-value of insulation alone ($m^2°C/W$)
Composite	0.261	3.5	0.409	2.1
Hot and Dry	0.261	3.5	0.409	2.1
Warm and Humid	0.261	3.5	0.409	2.1
Moderate	0.409	2.1	0.409	2.1
Cold	0.261	3.5	0.409	2.1

Table 10.3: Vertical Fenestration U-factor and SHGC Requirements (U-factor in $W/m^2 °C$)

		WWR≤40%	40% <WWR≤60%
Climate	Maximum U-factor	Maximum SHGC	Maximum SHGC
Composite	3.30	0.25	0.20
Hot and Dry	3.30	0.25	0.20
Warm and Humid	3.30	0.25	0.20
Moderate	6.90	0.40	0.30
Cold	3.30	0.51	0.51

Compliance Approaches

After establishing the specific climate zone in which the building is located, determine which compliance approach is the best fit for envelope design. The ECBC allows the following approaches:

Prescriptive Approach: prescribes the minimum performance requirements for each building component. It is quick and easy to use, but this approach is somewhat restrictive because requirements have to be met exactly as specified.

The prescriptive requirements for walls, roof and fenestrations are climate-based and different for buildings used during the daytime, and those with 24-hours of use. Roofs and Opaque Walls should meet maximum U-factors for assemblies or minimum R-values for the insulation only [ECBC 4.3.1 and 4.3.2].

Cool Roof should meet minimum solar reflectance of 0.7 and initial emittance levels of not less than 0.75, and determined in accordance with specified standards. [ECBC 4.3.1.1]

Skylights should meet maximum U-factor and SHGC with skylight area limited to a maximum of five percent of the gross roof area [ECBC 4.3.4].

Envelope Trade-off Approach: allows the designer to trade enhanced energy efficiency in one building component against decreased energy efficiency in another component, thereby offering flexibility.

These trade-offs are applicable only within major envelope components i.e., roof, walls and fenestration.

Appendix D of ECBC provides guidance on the calculation of the Envelope Performance Factor (EPF). This is calculated for the proposed design and for the baseline design, for compliance. The proposed building's EPF must be equal to or better than that of the baseline design.

Whole Building Performance: Approach compares the proposed design with a standard design (same building meeting all the prescriptive requirements of ECBC) and demonstrates that the estimated annual energy use of proposed design is less than that of the standard design. This approach allows great flexibility but requires considerably more effort. Computer-based Energy Simulation Program can be helpful in this exercise.

This approach (Appendix B of ECBC) requires computer-based energy simulation program to determine and compare the estimated annual energy use of the proposed design with that of a standard design.

10.6 ECBC Guidelines on Heating Ventilation and Air conditioning System

Heating ventilation and air conditioning (HVAC) refers to the equipment, distribution systems and terminals that provide HVAC requirement of the building. The best HVAC design considers all the interrelated building systems while addressing indoor air quality, thermal comfort, energy consumption and environmental benefits.

The Energy Conservation Building Code (ECBC) covers several prescriptive requirements for HVAC systems including requirements for economizer, duct and pipe insulation, controls optimization, and system balancing.

Equipment Efficiency

HVAC equipment is required to meet or exceed minimum efficiency requirements mentioned in ECBC 5.2.2. Power consumption ratings for unitary air conditioners, split systems and packaged air conditioners are referred to BIS codes. Cooling systems not included are referred to ASHRAE 90.1 – 2004. Single zone unitary systems are covered as well as multiple zone air and water systems. The more complex the system, the more requirements apply to that system: a single-zone unitary system has fewer requirements than a complex system made up of chillers, boilers, and fan coil units. For natural ventilation requirements, buildings are required to follow the design guidelines provided for natural ventilation in the National Building Code of India, 2005 [ECBC 5.2.1].

Controls

Controls determine how HVAC systems operate to meet the design goals of comfort, efficiency, and cost-effective operation. The ECBC requires that all HVAC equipment (>28 kW cooling and/or >7 kW heating capacity) be controlled with a time clock capable of retaining programming for at least 10 hours, controlling varying schedules. [ECBC 5.2.3]

All heating and cooling systems are required to be temperature controlled [ECBC 5.2.3]. Each zone must be controlled by an individual temperature controller. A temperature of dead band of 3ºC (5ºF) is required for equipment that supplies both heating and cooling. Thermostats must also prevent simultaneous heating and cooling. [ECBC 5.2.3.2].

Distribution System

Distribution systems carry a heating or cooling medium to condition the space. The two most common mediums are air and water. Air-based systems are often "forced air" because they use a fan to push the air from the furnace or air conditioner through the duct work to the conditioned space. Water-based systems are often called hydronic, and steam based systems are called steam systems, both of which use a piping system to circulate water or steam.

Forced hot air systems are more common than hydronic or steam heating systems because they cost much less to install. However, hydronic systems perform better than forced hot air because they leak less heat, are easier to insulate, make almost no noise, can be easily zoned and can provide more even temperatures. Hydronic systems also do a much better job of serving a radiant heating system. The energy efficiency of a distribution system depends largely on the design and installation quality. Duct distribution systems are prone to the most significant losses - especially if the ducts are poorly sealed and/or installed outside the "thermal envelope" of the building (in an unconditioned attic, for example). Hydronic systems are typically installed within conditioned or "buffered" spaces like an unconditioned basement. In either case, it is important to insulate ducts/ hot water pipes that are in unconditioned spaces.

The ECBC requires insulating ducts and pipelines to reduce energy losses in heating and cooling distribution systems. Insulation exposed to weather is required to be protected by aluminum sheet metal, painted canvas, or plastic cover. Cellular foam insulation needs to be protected as described above, or be painted with water retardant paint.

Duct sealing: Duct sealing applies to supply and return duct work and to plenums that are formed by part of the building envelope. Proper duct sealing ensures that correct quantities of heated or cooled air is delivered to the space, and not be lost to unconditioned spaces or the outdoors through leaks in the ducts. This may be one of the most important conservation features to check. A properly sealed duct system will increase the comfort and lower the energy use of the building. Some areas to be sealed include:

- Longitudinal seams are joints oriented in the direction of air flow.
- Transverse joints are connections of two duct sections orientated perpendicular to airflow.
- Duct wall penetrations are openings made by any screw or fastener.
- Spiral lock joints in round and flat oval duct need not be sealed.

Pipe insulation: To minimize heat losses, the Code requires that piping of heating and cooling systems, (including the storage tanks,) must be insulated. The Code specifies required R-values of insulation for heating and cooling systems based on the operating temperature of the system.

Air System Balancing

Air balancing is a small part of the building commissioning process. Commissioning involves the functional testing of all components of an HVAC system to insure proper operation. Commissioning is a systematic process of verification and documentation from the design phase to a minimum of one year after the construction that all systems perform in accordance with design documentation and intent, and in accordance with the operational needs. The air-balancing portion of the commissioning process is usually done at the completion of a new construction project.

Air balancing should verify damper operation and adjust settings to deliver the designed airflow to each zone. It is important that some balancing be done prior to occupancy for several reasons. Some equipment has minimum airflow requirements across the coils to avoid compressor damage. The total supply, return, and outside airflow quantities should be measured for each air handling system. The ECBC requires that air systems be balanced in a manner to first minimize throttling losses; then for fans > 0.75 kW (1.0 HP) the fan speed needs to be adjusted to meet design flow conditions. [ECBC 5.2.5.1.1].

Hydronic System Balancing

A balanced hydronic system is one that delivers even flow to all the devices on that piping system. Each component has an effective equal length of pipe on the supply and return. And when a system is balanced, all of the pressure drops are correct for the devices. When that happens, the highest efficiencies are possible in the system, which translates into reduced pumping costs.

ECBC requires hydronic systems to be proportionately balanced in a manner to first minimize throttling losses; then the pump impeller must be trimmed or pump speed adjusted to meet the design flow conditions [ECBC 5.2.5.1.2].

The ECBC addresses Variable Flow Hydronic Systems [ECBC 5.3.2] in several ways. First, it requires that pump flow rates be controllable to either 50% of the design or the minimum required by the equipment manufacturer for proper operation of the chillers or boilers. Second, the ECBC requires that circulation pumps ≥ 3.7 kW (5 HP) in water-cooled air-conditioning units or heat pumps contain a two-way automatic isolation valve to shut off the condenser water flow when the compressor is not operating [ECBC 5.3.2.2]. Thirdly, the ECBC requires pump motors in all hydronic systems ≥ 3.7 kW (5 HP) shall be controlled with variable speed drives [ECBC 5.3.2.3]. While throttling reduces the flow, the motor is still running at full speed and works even harder as it has to work against a restriction. By reducing the speed of the motor, the variable speed drive ensures reduction in energy consumption while maintaining the required flow.

Scale Control in Water Circuit

In a water-cooled air-conditioning system, heat is rejected from the refrigerant to the cooling water in the condenser. The impurities in the cooling water circuit get accumulated, and thus the scales and deposits are built up in the condenser tubes, creating scaling problems on the condenser heat transfer surfaces. This reduces the heat transfer efficiency of the condenser and thus increases chiller energy consumption. The ECBC requires [ECBC 5.2.6.2] use of soft water for condensers and chilled water systems to reduce scale formation.

Economizers

An economizer is a collection of dampers, sensors, actuators, and logic devices that together decide how much outside air to bring into a building. When the outdoor temperature and humidity are mild, economizers save energy by cooling buildings with outside air instead of by using refrigeration equipment to cool recirculated air. A properly operating economizer can cut energy costs by as much as 10 percent of a building's total energy consumption, depending mostly on local climate and internal

cooling loads. ECBC requires an economizer for cooling systems over 1,200 liters/sec (2,500 cfm) fan capacity and with a cooling capacity > 22 kW [ECBC 5.3.1.1].

10.7 ECBC Guidelines on Service Hot Water

Solar Water Heating

Residential facilities, hotels and hospitals with a centralized system shall have solar water heating for at least 20% of the design capacity.

Supplementary Water Heating System

Supplementary heating system shall be designed to maximize the energy efficiency of the system and shall incorporate the following design features in cascade:

a. Maximum heat recovery from hot discharge system like condensers of air conditioning units,

b. Use of gas fired heaters wherever gas is available, and

c. Electric heater as last resort.

Piping Insulation

The entire hot water system including the storage tanks, pipelines shall be insulated conforming to the relevant IS standards on materials and applications.

Heat Traps

Vertical pipe risers serving storage water heaters and storage tanks not having integral heat traps and serving a non-recirculating system shall have heat traps on both the inlet and outlet piping as close as practical to the storage tank.

Swimming Pools

Heated pools shall be provided with a vapor retardant pool cover on or at the water surface. Pools heated to more than 32°C (90°F) shall have a pool cover with a minimum insulation value of R-2.1 (R-12).

10.8 ECBC Guidelines on Lighting

Lighting systems shall apply to:

a. Interior spaces of buildings,

b. Exterior building features, including facades, illuminated roofs, architectural features, entrances, exits, loading docks, and illuminated canopies, and,

c. Exterior building grounds lighting that is provided through the building's electrical service.

Lighting Control

Interior lighting systems in buildings larger than 500 m^2 (5,000 ft^2) shall be equipped with an automatic control device. Within these buildings, all office areas less than 30 m2 (300 ft^2) enclosed by walls or ceiling-height partitions, all meeting and conference rooms, all school classrooms, and all storage spaces shall be equipped with occupancy sensors. For other spaces, this automatic control device shall function on either

a) A scheduled basis at specific programmed times. An independent program schedule shall be provided for areas of no more than 2,500 m^2 (25,000 ft^2) and not more than one floor; or,

b) Occupancy sensors that shall turn the lighting off within 30 minutes of an occupant leaving the space. Light fixtures controlled by occupancy sensors shall have a wall mounted, manual switch capable of turning off lights when the space is occupied.

Lighting for all exterior applications shall be controlled by a photo sensor or astronomical time switch that is capable of automatically turning off the exterior lighting when daylight is available or the lighting is not required.

Lighting Power Density (LPD)

The interior lighting power required for different activities are per ECBC guidelines are discussed. Lighting power density is an indicator of Power in watts per Sq meter. The lighting Power Density is derived by two methods namely:

- Building area method
- Space function method.

The installed interior lighting power calculated shall include all power used by the luminaries, including lamps, ballasts, current regulators, and control devices.

Building Area Method

Determination of interior lighting power allowance (watts) by the building area method (type of building) shall be in accordance with the LPD values given in code. The interior lighting power allowance is the sum of the products of the gross lighted floor area of each building area times the allowed lighting power density for that building area types.

Example: A hotel has four floors each of 1000 m^2 area. If the LPD is 10.8 W/m^2, find out the interior lighting power allowance.

= (1000 x 10.8) x 4

= 43200 W

Space Function Method

Determination of interior lighting power allowance (watts) by the space function method (type of operation) shall be in accordance with the LPD values provided in the code. The interior lighting power

allowance is the sum of the lighting power allowances for all spaces. The lighting power allowance for a space is the product of the gross lighted floor area of the space times the allowed lighting power density for that space.

Example: An enclosed office has 400 m^2 area. If the LPD is 11.8 W/m^2, find out the interior lighting power allowance.

= 400 X 11.8

= 4720 W

Exit Signs

Internally-illuminated exit signs shall not exceed 5 W per face.

Exterior Building Grounds Lighting

Lighting for exterior building grounds luminaires which operate at greater than 100 W shall contain lamps having a minimum efficacy of 60 lm/W unless the luminaire is controlled by a motion sensor.

For building exterior lighting applications specified in ECBC [Table 7.4], the connected lighting power shall not exceed the specified lighting power limits specified for each of these applications.

10.9 ECBC Guidelines on Electrical Power

ECBC has only mandatory requirements for electric power systems installed in buildings. These provisions are related to distribution transformers, electric motors, power factor, and distribution losses.

The mandatory requirements of ECBC cover the following electrical equipment and systems of building:

- Transformers
- Energy- Efficient Motors
- Power Factor Correction
- Electrical Metering and Monitoring
- Power Distribution Systems

Power transformers of the proper ratings and design must be selected to satisfy the minimum acceptable efficiency at 50% and full load rating. In addition, the transformer must be selected such that it minimizes the total of its initial cost in addition to the present value of the cost of its total lost energy while serving its estimated loads during its respective life span.

All permanently wired polyphase motors of 0.375 kW or more serving the building and expected to operate more than 1,500 hours per year and all permanently wired polyphase motors of 50kW or more serving the building and expected to operate more than 500 hours per year shall have a minimum acceptable nominal full load motor efficiency not less than IS 12615 for energy efficient motors. Motor horsepower ratings shall not exceed 200% of the calculated maximum load being served.

All electricity supplies exceeding 100 A, 3 phase shall maintain their power factor between 0.95 lag and unity at the point of connection.

Services exceeding 1000 kVA shall have permanently installed electrical metering to record demand (kVA), energy (kWh), and total power factor. The metering shall also display current (in each phase and the neutral), voltage (between phases and between each phase and neutral), and total harmonic distortion (THD) as a percentage of total current.

Services not exceeding 1000 kVA but over 65 kVA shall have permanently installed electric metering to record demand (kW), energy (kWh), and total power factor (or kVARh).

Services not exceeding 65 kVA shall have permanently installed electrical metering to record energy (kWh).

The power cabling shall be adequately sized as to maintain the distribution losses not to exceed 1% of the total power usage.

10.10 Building Water Pumping Systems

The pumps used in the buildings are for chilled water circulation, cooling water circulation, domestic water, hot water and sewage water. The water requirement for cooling, drinking and general purpose requirements in buildings are meat by the local authorities and also from the independent bore wells. The water received is stored and pumped to the overhead tanks provided on the building terrace are hydro pneumatic system is used. Normally the pumps used in the building are of centrifugal type with efficiencies of 60%. The following energy conservation measures are adopted to reduce energy consumption in water pumping in buildings:

- Installation of high efficiency pumps
- Operation of pumps in parallel
- Auto pump operation with low level/high level flow control systems.
- Installation of variable frequency drive for chiller water and cooling water pumps

10.11 Uninterruptible power supply

An Uninterruptible Power Supply (UPS) is a device that has an alternate source of energy, typically a battery backup that can provide power when the primary power source is temporarily disabled. The switchover time must be small enough to not cause a disruption in the operation of the loads (Figure 10.6).

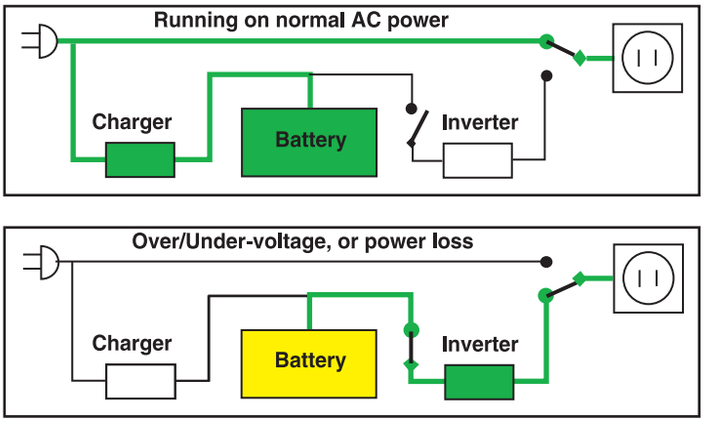


Figure 10.6 UPS Operation

The components of a UPS are converter (AC to DC), battery, inverter (DC to AC), monitor and control hardware / software.

There are two type of UPS architecture namely line interactive (OFF line) and ON line.

Simple OFF Line UPS systems, connect the load directly to the input AC line. Line Interactive systems have also the ability to correct UPS output if AC input voltage deviates beyond preset limits by means

of an auto- transformer based Automatic Voltage Regulator. In case of significant utility voltage deeps or outages these systems transfer the UPS to battery operation.

Due to its direct connection to input AC power, OFF Line types, including Line Interactive systems offer higher efficiency when compared to an Online UPS. But, unstable grid environment, with frequent power interruptions or outages, might cause these systems to suffer from frequent transfers to battery operation. Thus, exposing the critical load to possible failures due to unsmooth or unsuccessful transfer, or to battery failure, because of numerous battery discharges, which decrease drastically battery life time.

An Online UPS system, frequently called Double conversion system, first converts the AC input voltage to stabilized DC, which is then converted back to AC, to feed continuously the critical load, with pure stabilized sinusoidal output, coming either from the input line via AC/DC converter, or from batteries in case of power failure.

Energy efficiency of a UPS is the difference between the amount of energy that goes into a UPS versus the amount of useful energy that comes out of the UPS and actually powers loads (Figure 10.7). In all UPS systems, some energy is lost as heat when it passes through the internal components of the UPS (including transformers, rectifiers, and inverters).

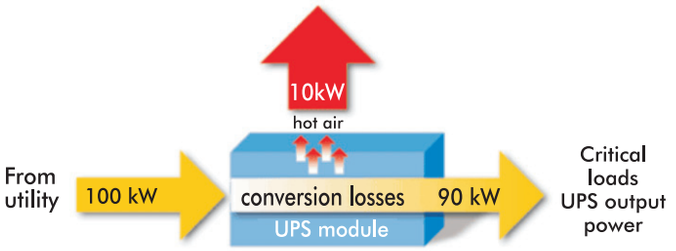


Figure 10.7 Energy Loss in UPS

Almost all UPS power 100% nonlinear loads +viz., computers, servers and electronic equipment, though the manufacturers test their UPS often using linear loads. UPS efficiency is often much higher when powering linear loads.

The second major factor to influence UPS efficiency is the power level at which the efficiency is measured. Most UPSs typically have their best efficiency operating at 50% to 100% load level. But in the real situation, most UPS systems operate at 25% to 60% of their nominal load- not fully loaded. To determine accurate efficiency, the UPS should demonstrate efficiency at loads between 25–50%, where most UPS will likely to be operating.

10.12 Escalators and Elevators

The requirement of the maximum allowable electric power indicates ultimately the energy performance of the equipment. The power for lift equipment is to be measured when the lift is carrying its rated load and moving upward at its contract speed. For escalators and passenger conveyors, since the rated load is usually defined as number of person (not in kg weight), there is no theoretical rated load in kg for the equipment. Thus the electric power is to be measured when the escalator/conveyor is carrying no load and moving at its rated speed either in the upward or downward direction.

In escalators and elevators, the dominating factors that determine the energy consumption are the efficiency of the motor, friction, the controller and the driving gear box. The proportion of frictional loss of the machine can also become significant in the power consumption in no load condition, as it is the fixed overhead to keep the equipment running.

Factors That Affect Energy Consumption in Elevators and Escalator System

Energy is consumed by lift and escalator equipment mainly on the following categories:

- ✓ Friction losses incurred while travelling.
- ✓ Dynamic losses while starting and stopping.
- ✓ Lifting (or lowering) work done, potential energy transfer.
- ✓ Regeneration into the supply system.

The general approach to energy efficiency in lift and escalator equipment is merely to minimize the friction losses and the dynamic losses of the system. There are many factors that will affect these losses for elevator and escalator system:-

(A) Characteristic of the equipment

The type of motor, drive and control system of the machine, the internal decoration, means to reduce friction in moving parts (e.g. guide shoes), type, speed and the pulley system of the equipment.

(B) Characteristic of the premises

The population distribution, the type of the premises, the height of the premises and the house keeping of the premises,

(C) The configuration of the lift/escalator system

The zoning of the lift system, the combination of lift and escalator equipment, the strategies for vertical transportation and the required grade of service of the system.

General Principles to Achieve Energy Efficiency

In general the principles for achieving energy efficiency for lift/escalator installations are as follows:

- ✓ Specify energy efficiency equipment for the system.
- ✓ Do not over design the system.
- ✓ Suitable zoning arrangement.
- ✓ Suitable control and energy management of lift equipment
- ✓ Use light weight materials for lift car decoration.
- ✓ Good house keeping.

10.13 Building Energy Management System (BEMS)

Energy management systems can vary considerably in complexity and degree of sophistication. The simplest timing mechanism to switch systems ON and OFF at pre-determined intervals on a routine basis could be considered as an energy management system. These progresses to include additional features such as programmers, thermostatic controls, motorised valves, zoning, and optimum start controllers and compensated circuits.

The most complex of energy management systems have a computerised central controller linked to numerous sensors and information sources. These could include the basic internal and external range shown schematically in Figure 10.8, along with further processed data to include: the time, the day of the week, time of year, percentage occupancy of a building, meteorological data, system state feedback factors for plant efficiency at any one time and energy gain data from the sun, lighting, machinery and people.

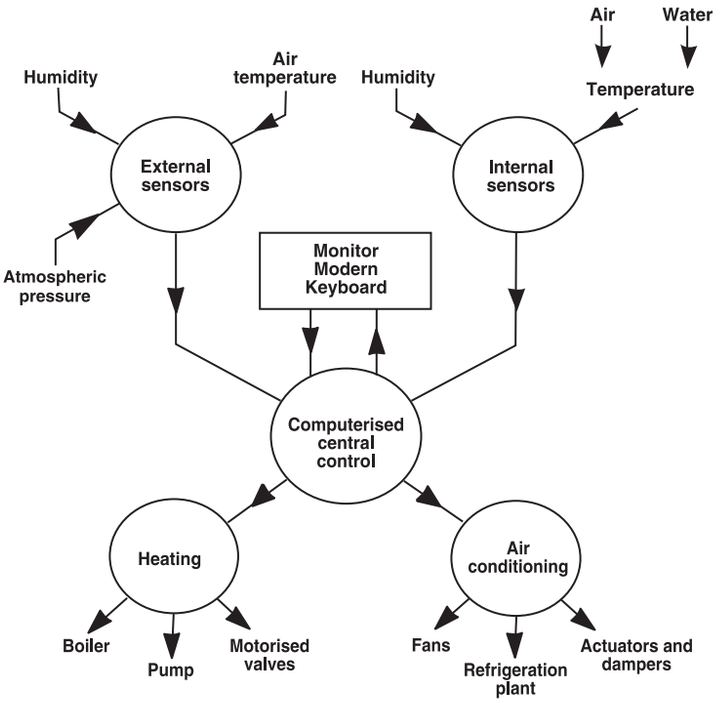


Figure 10.8 Building Energy Management

A microprocessor is then the main feature of the control system. Data on temperature, flow rates, pressures, etc., as appropriate, are collected from sensors in the system and the treated spaces and stored in the memory of the processor. Provided that equations defining the performance of the control elements, the items of plant and the behavioural characteristics of the systems controlled have been developed and fed into the micro-processor as algorithms, deviations from the desired performance can be dealt with by calculation, the plant output being varied accordingly. Mathematical functions replace control modes. For example, if room temperature rose in a space conditioned by a constant volume reheat system, the correct position of the valve in the low temperature hot water line feeding the heater battery could be calculated and corrected as necessary, to bring the room temperature back to the set point as rapidly as possible, without any offset. Data can be stored to establish trends and anticipation can be built into the program so that excessive swings in controlled conditions may be prevented. Furthermore, self-correction can be incorporated so that the control system learns from experience and the best possible system performance is obtained. This implies that commissioning inadequacies and possibly even design faults can be corrected but only to a certain extent. Optimum results are only obtainable, and the cost of the installation justified, from systems that have been properly designed, installed and commissioned. Under such circumstances it is then feasible to extend the scope of microprocessor control to include the management of all the building services with an economic use of its thermal and electrical energy needs.

The functions of a building management system (BMS) or building energy management system (BEMS) are monitoring and control of the services and functions of a building, in a way that is economical and efficient in the use of energy. Furthermore, it may be arranged that one system can control a group of buildings.

10.14 Star Rating of Buildings

Star rating programme for buildings is based on the actual performance of a building in terms of its specific energy usage in kwh/sqm/year. The programme rates office buildings on a 1-5 Star scale, with

5 Star labeled buildings being the most efficient. The label provided under it is applicable for a period of 5 years from the date of issue. The Star rating Programme provides public recognition to energy efficient buildings. Different categories of buildings such as office buildings (day use and BPOs), Shopping Malls, Hotels, Hospitals and IT parks in the five climatic zones have been labeled under the scheme.

The rating normalizes for operational characteristics that define the building use, hours of operation, climatic zone and conditioned space. To provide a useful benchmark the rating provides a comparison to the building's peer group representing those buildings that have the same primary business function, and operating characteristics. The rating is based on an analysis of national data that accurately reflects the distribution of energy use for each building type.

The national energy performance rating is a type of external benchmark that helps energy managers to assess how efficiently their buildings use energy, relative to similar buildings nationwide. Additionally, building owners and managers can use the performance ratings to help identify buildings that offer the best opportunity for improvement and recognition.

The objective of the Star rating is to create a market pull for efficient buildings. Improved energy efficiency is a primary goal which could be achieved through promotion of higher performance in buildings. Building rating and verification systems are an effective measure to encourage building owners to go beyond the minimum and creating an awareness of these systems would add substantial momentum to promote energy efficiency.

Energy performance Index: (EPI)

Energy Performance Index (EPI) in kWh / sq m/ year will be considered for rating the building. Bandwidths for Energy Performance Index for different climatic zones have been developed based on percentage air-conditioned space. For example a building in a composite climatic zone like New Delhi and having air conditioned area greater than 50% of their built up area, the bandwidths of EPI range between 190-90 kWh/sq m/year. Thus a building would get a 5-Star rating if its EPI falls below 90 kWh/sq m/year and 1 Star if it is between 165-190 kWh/sq m/year.

In addition to the star rating of the existing office buildings a Specific Star Rating programme has also been introduced for Business process outsourcing buildings such as call centers and allied services. The salient features of the BPO star rating programmes is highlighted below.

Average Annual hourly Energy Performance Index (EPI) i.e. (AAhEPI) in (Wh/sqm)/hr will be considered for rating the BPO building. AAhEPI shall be in terms of purchased and generated electricity divided by built up area in sqm and total annual hours of operation. However the total electricity would not include electricity generated from on-site renewable sources such as solar photovoltaic etc.

AAhEPI – AAhEPI (Wh/sqm)/hr is defined as Average Annual hourly EPI of a Building. This value is calculated from EPI values, mentioned in the table 10.12, as follows

[EPI/ (Daily hours of Operation X days of operation in a week X 52 weeks in a year)] X 1000. This value will be independent of time and therefore applicable to all types of BPO operations.

Built up Area – Built up area is the carpet area + area covered by thickness of walls + balconies etc. and would exclude basement and other areas used for parking. In case basement is being utilized for purpose other than parking, it should be included.

10.15 Energy Efficiency Measures in Buildings:

Air-Conditioning System:

Weather stripping of Windows and Doors

Minimise exfiltration of conditioned air and infiltration of external un-conditioned air through leaky windows and doors by incorporating effective means of weather stripping. Self-closing doors should also be provided where heavy traffic of people is anticipated.

Temperature and Humidity Setting

Ensure human comfort by setting the temperature to between 23°C and 25°C and the relative humidity between 55% to 65%.

Chilled Water Leaving Temperature

Ensure higher chiller energy efficiency by maintaining the chilled water leaving temperature at or above 7° C. As a rule of thumb, the efficiency of a centrifugal chiller increases by about 2¼ % for every 1° C rise in the chilled water leaving temperature.

Chilled Water Pipes and Air Ducts

Ensure that the insulation for the chilled water pipes and ducting system is maintained in good condition. This helps to prevent heat gain from the surroundings.

Chiller Condenser Tubes

Ensure that mechanical cleaning of the tubes is carried out at least once every six months. Fouling in the condenser tubes in the form of slime and scales reduces the heat transfer of the condenser tubes and thereby reducing the energy efficiency of the chiller.

Cooling Towers

Ensure that the cooling towers are clean to allow for maximum heat transfer so that the temperature of the water returning to the condenser is less than or equal to the ambient temperature.

Air-handling Unit Fan Speed

Install devices such as frequency converters to vary the fan speed. This will reduce the energy consumption of the fan motor by as much as 15%.

Air Filter Condition

Maintain the filter in a clean condition. This will improve the heat transfer between air and chilled water and correspondingly reduce the energy consumption.

Lighting System:

All lighting systems generate heat that needs to be dissipated. By designing energy efficient lighting system that integrates day lighting and good controls, heat gains can be reduced significantly. This can reduce the size of the HVAC system resulting in first-cost savings.

Day lighting

Day lighting benefits go beyond energy savings and power reduction. Daylight spaces have been shown to improve people's ability to perform visual tasks, increase productivity and reduce illness. Building fenestration should be designed to optimize day lighting and reduce the need for electric lighting. Orient the building to minimize building exposure to the east and west and maximize glazing on the south and north exposures.

Daylight strategies do not save energy unless electric lights are turned off or dimmed appropriately. ECBC requires controls in day lit areas that are capable of reducing the light output from luminaires by at least half.

- Install dimmers to take advantage of day lighting and where cost-effective.
- Replace rheostat dimmers with efficient electronic dimmers.
- Combine time switching with day lighting using astronomical time clocks.
- Control exterior lighting with photo controls where lighting can be turned off after a fixed interval.

Switch off Lights When Not in Use

Provision of Separate Switches for Peripheral Lighting

A flexible lighting system, which made use of natural lighting for the peripherals of the room, should be considered so that these peripheral lights can be switched off when not needed.

Install High Efficiency Lighting System

Replace incandescent and other inefficient lamps with lamps with higher lighting efficacy. For example, replacing incandescent bulbs with compact fluorescent lamps can reduce electricity consumption by 75% without any reduction in illumination levels.

Fluorescent Tube Ballasts

The ballast losses of conventional ballast and electronic ballast are 12W and 2W respectively. Hence, consider the use of electronic ballast for substantial energy savings in the lighting system.

Lamp Fixtures or Luminaries

Optical lamp luminaries made of aluminum, silver or multiple dielectric coatings have better light distribution characteristics. Use them to reduce electricity consumption by as much as 50% without compromising on illumination levels.

Integration of Lighting System with Air-Conditioning System

In open plan offices, the air-conditioning and lighting systems can be combined in such a way that the return air is extracted through the lighting luminaires. This measure ensures that lesser heat will be directed from the lights into the room.

Cleaning of Lights and Fixtures

Clean the lights and fixtures regularly. For best results, dust at least four times a year.

Use Light Colors for Walls, Floors and Ceilings

The higher surface reflectance values of light colors will help to make the most of any existing lighting system. Consider light colored furniture and room partitions to optimize light reflectance. Avoid furniture colors and placement that will interfere with light distribution. Keep ceilings and walls as bright as possible.

Deal with each activity area and each fixture individually

Eliminate excessive lighting by reducing the total lamp wattage in each activity area

Task Lighting

Lighting layout should use task lighting principle. Install focusing lamps or flexible extensions wherever needed.

QUESTIONS	
Objective Type Questions	
1.	EC Act covers buildings having connected load of a) 500 kW and above b) 500 kVA and above c) 100 kW and above d) all HT connection buildings
2.	LPD in ECBC refers to a) low power density b) Lux power density c) Lighting power density d) Low power demand
3.	Energy performance index (EPI) kWh/m^2 /yr is calculated based on a) total building annual energy consumption/ built up area b) total building annual energy consumption/ Carpet area c) total building annual energy consumption for HVAC & lighting/ built up area d) none of the above
4.	Which of the following is not a climate zone as per ECBC classification? a) hot-dry b) warm-humid c) cold d) cold humid
5.	A cool roof a) absorbs heat b) reflects heat c) transmits heat d) conducts heat
6.	SHGC in building envelope energy analysis is a) solar heat gain coefficient b) solar hot water gain coefficient c) standard heat gain coefficient d) standard hot glass coefficient
7.	Which one is not a fenestration a) windows b) skylights c) ventilators d) valves
8.	The ratio of daylight passing through the glazing to light passing through perfectly transmissive glazing is called a) visible light transmittance b) SHGC c) luminance index d) glazing coefficient
9.	The unit of AAhEPI is a) kWh/m^2 /yr b) m^2 x kWh/hr c) (Wh/sqm)/hr d) m^2 /Wh/yr
10.	In a UPS the conversion from DC to AC is carried out by a) converter b) charger c) battery d) inverter
Short Type Questions	
S-1	Name three applications of VFD's in a buildings
S-2	What do you understand by building envelope? What are its functions?
S-3	Explain briefly the meaning of solar heat gain coefficient

S-4	What is the significance of overall heat transfer coefficient (U-factor) in building energy consumption?
S-5	Explain the term "Building Management System (BMS)"
Long Type Questions	
L-1	List ten energy consumption measures in buildings?
L-2	Explain the following terms a) solar reflectance b) fenestration c) visible light transmittance d) effective aperture of glazing

REFERENCES

1. ECBC Guide Book
2. Star Rating Programme for Existing office building
3. Star Rating Programme for BPO buildings
4. NPC Case studies

ANNEXURE

CHECKLIST & TIPS FOR ENERGY EFFICIENCY IN ELECTRICAL UTILITIES

Electricity

- Optimise the tariff structure with utility supplier
- Schedule your operations to maintain a high load factor
- Shift loads to off-peak times if possible.
- Minimise maximum demand by tripping loads through a demand controller
- Stagger start-up times for equipment with large starting currents to minimize load peaking.
- Use standby electric generation equipment for on-peak high load periods.
- Correct power factor to at least 0.90 under rated load conditions.
- Relocate transformers close to main loads.
- Set transformer taps to optimum settings.
- Disconnect primary power to transformers that do not serve any active loads
- Consider on-site electric generation or cogeneration.
- Export power to grid if you have any surplus in your captive generation
- Check utility electric meter with your own meter.
- Shut off unnecessary computers, printers, and copiers at night.

Motors

- Properly size to the load for optimum efficiency.
  (High efficiency motors offer of 4 - 5% higher efficiency than standard motors)
- Use energy-efficient motors where economical.
- Use synchronous motors to improve power factor.
- Check alignment.
- Provide proper ventilation
  (For every 10^{o}C increase in motor operating temperature over recommended peak, the motor life is estimated to be halved)
- Check for under-voltage and over-voltage conditions.
- Balance the three-phase power supply.
  (An Imbalanced voltage can reduce 3 - 5% in motor input power)
- Demand efficiency restoration after motor rewinding.
  (If rewinding is not done properly, the efficiency can be reduced by 5 - 8%)

Drives

- Use variable-speed drives for large variable loads.
- Use high-efficiency gear sets.
- Use precision alignment.
- Check belt tension regularly.
- Eliminate variable-pitch pulleys.
- Use flat belts as alternatives to v-belts.
- Use synthetic lubricants for large gearboxes.

- Eliminate eddy current couplings.
- Shut them off when not needed.

Fans

- Use smooth, well-rounded air inlet cones for fan air intakes.
- Avoid poor flow distribution at the fan inlet.
- Minimize fan inlet and outlet obstructions.
- Clean screens, filters, and fan blades regularly.
- Use aerofoil-shaped fan blades.
- Minimize fan speed.
- Use low-slip or flat belts.
- Check belt tension regularly.
- Eliminate variable pitch pulleys.
- Use variable speed drives for large variable fan loads.
- Use energy-efficient motors for continuous or near-continuous operation
- Eliminate leaks in ductwork.
- Minimise bends in ductwork
- Turn fans off when not needed.

Blowers

- Use smooth, well-rounded air inlet ducts or cones for air intakes.
- Minimize blower inlet and outlet obstructions.
- Clean screens and filters regularly.
- Minimize blower speed.
- Use low-slip or no-slip belts.
- Check belt tension regularly.
- Eliminate variable pitch pulleys.
- Use variable speed drives for large variable blower loads.
- Use energy-efficient motors for continuous or near-continuous operation.
- Eliminate ductwork leaks.
- Turn blowers off when they are not needed.

Pumps

- Operate pumping near best efficiency point.
- Modify pumping to minimize throttling.
- Adapt to wide load variation with variable speed drives or sequenced control of smaller units.
- Stop running both pumps -- add an auto-start for an on-line spare or add a booster pump in the problem area.
- Use booster pumps for small loads requiring higher pressures.
- Increase fluid temperature differentials to reduce pumping rates.
- Repair seals and packing to minimize water waste.
- Balance the system to minimize flows and reduce pump power requirements.
- Use siphon effect to advantage: don't waste pumping head with a free-fall (gravity) return.

Compressors

- Consider variable speed drive for variable load on positive displacement compressors.
- Use a synthetic lubricant if the compressor manufacturer permits it.
- Be sure lubricating oil temperature is not too high (oil degradation and lowered viscosity) and not too low (condensation contamination).
- Change the oil filter regularly.
- Periodically inspect compressor intercoolers for proper functioning.
- Use waste heat from a very large compressor to power an absorption chiller or preheat process or utility feeds.
- Establish a compressor efficiency-maintenance program. Start with an energy audit and follow-up, then make a compressor efficiency-maintenance program a part of your continuous energy management program.

Compressed air

- Install a control system to coordinate multiple air compressors.
- Study part-load characteristics and cycling costs to determine the most-efficient mode for operating multiple air compressors.
- Avoid over sizing -- match the connected load.
- Load up modulation-controlled air compressors. (They use almost as much power at partial load as at full load.)
- Turn off the back-up air compressor until it is needed.
- Reduce air compressor discharge pressure to the lowest acceptable setting.
  (Reduction of 1 kg/cm^2 air pressure (8 kg/cm^2 to 7 kg/cm^2) would result in 9% input power savings. This will also reduce compressed air leakage rates by 10%)
- Use the highest reasonable dryer dew point settings.
- Turn off refrigerated and heated air dryers when the air compressors are off.
- Use a control system to minimize heatless desiccant dryer purging.
- Minimize purges, leaks, excessive pressure drops, and condensation accumulation.
  (Compressed air leak from 1 mm hole size at 7 kg/cm^2 pressure would mean power loss equivalent to 0.5 kW)
- Use drain controls instead of continuous air bleeds through the drains.
- Consider engine-driven or steam-driven air compression to reduce electrical demand charges.
- Replace standard v-belts with high-efficiency flat belts as the old v-belts wear out.
- Use a small air compressor when major production load is off.
- Take air compressor intake air from the coolest (but not air conditioned) location.
  (Every 5^{0}C reduction in intake air temperature would result in 1% reduction in compressor power consumption)
- Use an air-cooled aftercooler to heat building makeup air in winter.
- Be sure that heat exchangers are not fouled (e.g. -- with oil).
- Be sure that air/oil separators are not fouled.
- Monitor pressure drops across suction and discharge filters and clean or replace filters promptly upon alarm.
- Use a properly sized compressed air storage receiver. Minimize disposal costs by using lubricant that is fully demulsible and an effective oil-water separator.

- Consider alternatives to compressed air such as blowers for cooling, hydraulic rather than air cylinders, electric rather than air actuators, and electronic rather than pneumatic controls.
- Use nozzles or venturi-type devices rather than blowing with open compressed air lines.
- Check for leaking drain valves on compressed air filter/regulator sets. Certain rubber-type valves may leak continuously after they age and crack.
- In dusty environments, control packaging lines with high-intensity photocell units instead of standard units with continuous air purging of lenses and reflectors.
- Establish a compressed air efficiency-maintenance program. Start with an energy audit and follow-up, then make a compressed air efficiency-maintenance program a part of your continuous energy management program.

Chillers

- Increase the chilled water temperature set point if possible.
- Use the lowest temperature condenser water available that the chiller can handle.
  (Reducing condensing temperature by 5.5^{0}C, results in a 20 - 25% decrease in compressor power consumption)
- Increase the evaporator temperature
  (5.5^{0}C increase in evaporator temperature reduces compressor power consumption by 20 - 25%)
- Clean heat exchangers when fouled.
  (1 mm scale build-up on condenser tubes can increase energy consumption by 40%)
- Optimize condenser water flow rate and refrigerated water flow rate.
- Replace old chillers or compressors with new higher-efficiency models.
- Use water-cooled rather than air-cooled chiller condensers.
- Use energy-efficient motors for continuous or near-continuous operation.
- Specify appropriate fouling factors for condensers.
- Do not overcharge oil.
- Install a control system to coordinate multiple chillers.
- Study part-load characteristics and cycling costs to determine the most-efficient mode for operating multiple chillers.
- Run the chillers with the lowest operating costs to serve base load.
- Avoid oversizing -- match the connected load.
- Isolate off-line chillers and cooling towers.
- Establish a chiller efficiency-maintenance program. Start with an energy audit and follow-up, then make a chiller efficiency-maintenance program a part of your continuous energy management program.

HVAC (Heating / Ventilation / Air Conditioning)

- Tune up the HVAC control system.
- Consider installing a building automation system (BAS) or energy management system (EMS) or restoring an out-of-service one.
- Balance the system to minimize flows and reduce blower/fan/pump power requirements.
- Eliminate or reduce reheat whenever possible.

- Use appropriate HVAC thermostat setback.
- Use morning pre-cooling in summer and pre-heating in winter (i.e. -- before electrical peak hours).
- Use building thermal lag to minimize HVAC equipment operating time.
- In winter during unoccupied periods, allow temperatures to fall as low as possible without freezing water lines or damaging stored materials.
- In summer during unoccupied periods, allow temperatures to rise as high as possible without damaging stored materials.
- Improve control and utilization of outside air.
- Use air-to-air heat exchangers to reduce energy requirements for heating and cooling of outside air.
- Reduce HVAC system operating hours (e.g. -- night, weekend).
- Optimize ventilation.
- Ventilate only when necessary. To allow some areas to be shut down when unoccupied, install dedicated HVAC systems on continuous loads (e.g. -- computer rooms).
- Provide dedicated outside air supply to kitchens, cleaning rooms, combustion equipment, etc. to avoid excessive exhausting of conditioned air.
- Use evaporative cooling in dry climates.
- Reduce humidification or dehumidification during unoccupied periods.
- Use atomization rather than steam for humidification where possible.
- Clean HVAC unit coils periodically and comb mashed fins.
- Upgrade filter banks to reduce pressure drop and thus lower fan power requirements.
- Check HVAC filters on a schedule (at least monthly) and clean/change if appropriate.
- Check pneumatic controls air compressors for proper operation, cycling, and maintenance.
- Isolate air conditioned loading dock areas and cool storage areas using high-speed doors or clear PVC strip curtains.
- Install ceiling fans to minimize thermal stratification in high-bay areas.
- Relocate air diffusers to optimum heights in areas with high ceilings.
- Consider reducing ceiling heights.
- Eliminate obstructions in front of radiators, baseboard heaters, etc.
- Check reflectors on infrared heaters for cleanliness and proper beam direction.
- Use professionally-designed industrial ventilation hoods for dust and vapor control.
- Use local infrared heat for personnel rather than heating the entire area.
- Use spot cooling and heating (e.g. -- use ceiling fans for personnel rather than cooling the entire area).
- Purchase only high-efficiency models for HVAC window units.
- Put HVAC window units on timer control.
- Don't oversize cooling units. (Oversized units will "short cycle" which results in poor humidity control.)
- Install multi-fueling capability and run with the cheapest fuel available at the time.
- Consider dedicated make-up air for exhaust hoods. (Why exhaust the air conditioning or heat if you don't need to?)
- Minimize HVAC fan speeds.

- Consider desiccant drying of outside air to reduce cooling requirements in humid climates.
- Consider ground source heat pumps.
- Seal leaky HVAC ductwork.
- Seal all leaks around coils.
- Repair loose or damaged flexible connections (including those under air handling units).
- Eliminate simultaneous heating and cooling during seasonal transition periods.
- Zone HVAC air and water systems to minimize energy use.
- Inspect, clean, lubricate, and adjust damper blades and linkages.
- Establish an HVAC efficiency-maintenance program. Start with an energy audit and follow-up, then make an HVAC efficiency-maintenance program a part of your continuous energy management program.

Refrigeration

- Use water-cooled condensers rather than air-cooled condensers.
- Challenge the need for refrigeration, particularly for old batch processes.
- Avoid oversizing -- match the connected load.
- Consider gas-powered refrigeration equipment to minimize electrical demand charges.
- Use "free cooling" to allow chiller shutdown in cold weather.
- Use refrigerated water loads in series if possible.
- Convert firewater or other tanks to thermal storage.
- Don't assume that the old way is still the best -- particularly for energy-intensive low temperature systems.
- Correct inappropriate brine or glycol concentration that adversely affects heat transfer and/or pumping energy. If it sweats, insulate it, but if it is corroding, replace it first.
- Make adjustments to minimize hot gas bypass operation.
- Inspect moisture/liquid indicators.
- Consider change of refrigerant type if it will improve efficiency.
- Check for correct refrigerant charge level.
- Inspect the purge for air and water leaks.
- Establish a refrigeration efficiency-maintenance program. Start with an energy audit and follow-up, then make a refrigeration efficiency-maintenance program a part of your continuous energy management program.

Cooling towers

- Control cooling tower fans based on leaving water temperatures.
- Control to the optimum water temperature as determined from cooling tower and chiller performance data.
- Use two-speed or variable-speed drives for cooling tower fan control if the fans are few. Stage the cooling tower fans with on-off control if there are many.
- Turn off unnecessary cooling tower fans when loads are reduced.
- Cover hot water basins (to minimize algae growth that contributes to fouling).
- Balance flow to cooling tower hot water basins.

- Periodically clean plugged cooling tower water distribution nozzles.
- Install new nozzles to obtain a more-uniform water pattern.
- Replace splash bars with self-extinguishing PVC cellular-film fill.
- On old counterflow cooling towers, replace old spray-type nozzles with new square-spray ABS practically-non-clogging nozzles.
- Replace slat-type drift eliminators with high-efficiency, low-pressure-drop, self-extinguishing, PVC cellular units.
- If possible, follow manufacturer's recommended clearances around cooling towers and relocate or modify structures, signs, fences, dumpsters, etc. that interfere with air intake or exhaust.
- Optimize cooling tower fan blade angle on a seasonal and/or load basis.
- Correct excessive and/or uneven fan blade tip clearance and poor fan balance.
- Use a velocity pressure recovery fan ring.
- Divert clean air-conditioned building exhaust to the cooling tower during hot weather.
- Re-line leaking cooling tower cold water basins.
- Check water overflow pipes for proper operating level.
- Optimize chemical use.
- Consider side stream water treatment.
- Restrict flows through large loads to design values.
- Shut off loads that are not in service.
- Take blowdown water from the return water header.
- Optimize blowdown flow rate.
- Automate blowdown to minimize it.
- Send blowdown to other uses (Remember, the blowdown does not have to be removed at the cooling tower. It can be removed anywhere in the piping system.)
- Implement a cooling tower winterization plan to minimize ice build-up.
- Install interlocks to prevent fan operation when there is no water flow.
- Establish a cooling tower efficiency-maintenance program. Start with an energy audit and follow-up, then make a cooling tower efficiency-maintenance program a part of your continuous energy management program.

Lighting

- Reduce excessive illumination levels to standard levels using switching, delamping, etc. (Know the electrical effects before doing delamping.)
- Aggressively control lighting with clock timers, delay timers, photocells, and/or occupancy sensors.
- Install efficient alternatives to incandescent lighting, mercury vapor lighting, etc. Efficiency (lumens/watt) of various technologies range from best to worst approximately as follows: low pressure sodium, high pressure sodium, metal halide, fluorescent, mercury vapor, incandescent.
- Select ballasts and lamps carefully with high power factor and long-term efficiency in mind.
- Upgrade obsolete fluorescent systems to Compact fluorescents and electronic ballasts
- Consider lowering the fixtures to enable using less of them.
- Consider daylighting, skylights, etc.

- Consider painting the walls a lighter color and using less lighting fixtures or lower wattages.
- Use task lighting and reduce background illumination.
- Re-evaluate exterior lighting strategy, type, and control. Control it aggressively.
- Change exit signs from incandescent to LED.

DG sets

- Optimise loading
- Use waste heat to generate steam/hot water /power an absorption chiller or preheat process or utility feeds.
- Use jacket and head cooling water for process needs
- Clean air filters regularly
- Insulate exhaust pipes to reduce DG set room temperatures
- Use cheaper heavy fuel oil for capacities more than 1MW

Buildings

- Seal exterior cracks/openings/gaps with caulk, gasketing, weatherstripping, etc.
- Consider new thermal doors, thermal windows, roofing insulation, etc.
- Install windbreaks near exterior doors.
- Replace single-pane glass with insulating glass.
- Consider covering some window and skylight areas with insulated wall panels inside the building.
- If visibility is not required but light is required, consider replacing exterior windows with insulated glass block.
- Consider tinted glass, reflective glass, coatings, awnings, overhangs, draperies, blinds, and shades for sunlit exterior windows.
- Use landscaping to advantage.
- Add vestibules or revolving doors to primary exterior personnel doors.
- Consider automatic doors, air curtains, strip doors, etc. at high-traffic passages between conditioned and non-conditioned spaces. Use self-closing doors if possible.
- Use intermediate doors in stairways and vertical passages to minimize building stack effect.
- Use dock seals at shipping and receiving doors.
- Bring cleaning personnel in during the working day or as soon after as possible to minimize lighting and HVAC costs.

Water & Wastewater

- Recycle water, particularly for uses with less-critical quality requirements.
- Recycle water, especially if sewer costs are based on water consumption.
- Balance closed systems to minimize flows and reduce pump power requirements.
- Eliminate once-through cooling with water.
- Use the least expensive type of water that will satisfy the requirement.
- Fix water leaks.
- Test for underground water leaks. (It's easy to do over a holiday shutdown.)

- Check water overflow pipes for proper operating level.
- Automate blowdown to minimize it.
- Provide proper tools for wash down -- especially self-closing nozzles.
- Install efficient irrigation.
- Reduce flows at water sampling stations.
- Eliminate continuous overflow at water tanks.
- Promptly repair leaking toilets and faucets.
- Use water restrictors on faucets, showers, etc.
- Use self-closing type faucets in restrooms.
- Use the lowest possible hot water temperature.
- Do not use a heating system hot water boiler to provide service hot water during the cooling season -- install a smaller, more-efficient system for the cooling season service hot water.
- If water must be heated electrically, consider accumulation in a large insulated storage tank to minimize heating at on-peak electric rates.
- Use multiple, distributed, small water heaters to minimize thermal losses in large piping systems.
- Use freeze protection valves rather than manual bleeding of lines.
- Consider leased and mobile water treatment systems, especially for deionized water.
- Seal sumps to prevent seepage inward from necessitating extra sump pump operation.
- Install pretreatment to reduce TOC and BOD surcharges.
- Verify the water meter readings. (You'd be amazed how long a meter reading can be estimated after the meter breaks or the meter pit fills with water!)
- Verify the sewer flows if the sewer bills are based on them

Miscellaneous

- Meter any unmetered utilities. Know what is normal efficient use. Track down causes of deviations.
- Shut down spare, idling, or unneeded equipment.
- Make sure that all of the utilities to redundant areas are turned off -- including utilities like compressed air and cooling water.
- Install automatic control to efficiently coordinate multiple air compressors, chillers, cooling tower cells, boilers, etc.
- Renegotiate utilities contracts to reflect current loads and variations.
- Consider buying utilities from neighbors, particularly to handle peaks.
- Leased space often has low-bid inefficient equipment. Consider upgrades if your lease will continue for several more years.
- Adjust fluid temperatures within acceptable limits to minimize undesirable heat transfer in long pipelines.
- Minimize use of flow bypasses and minimize bypass flow rates.
- Provide restriction orifices in purges (nitrogen, steam, etc.).
- Eliminate unnecessary flow measurement orifices.
- Consider alternatives to high pressure drops across valves.
- Turn off winter heat tracing that is on in summer.